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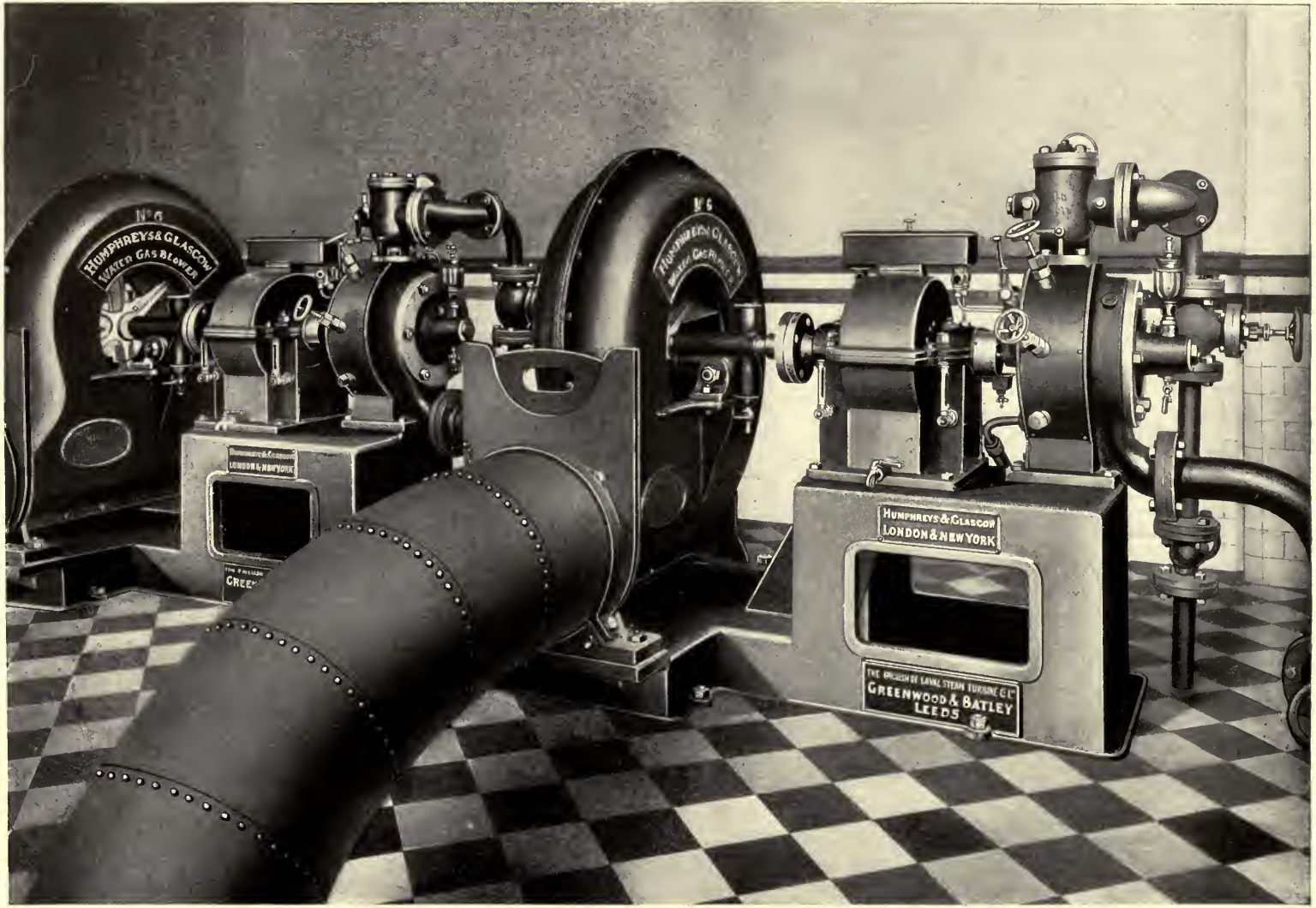
MODERN ENGINES
AND
POWER GENERATORS



Frontispiece.] TURBINE AIR PROPELLER.

MODERN ENGINES

AND

POWER GENERATORS

A PRACTICAL WORK ON PRIME MOVERS AND THE TRANSMISSION OF POWER STEAM, ELECTRIC, WATER, AND HOT AIR

BY

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"ELECTRICAL INSTALLATIONS" "ELECTRICAL DISTRIBUTION BY ALTERNATING CURRENTS AND TRANSFORMERS" "PHOTOGRAPHIC AND OPTICAL ELECTRIC LAMPS" AND NUMEROUS SCIENTIFIC ARTICLES AND PAPERS ON MECHANICAL AND ELECTRICAL ENGINEERING

WITH MANY HUNDRED ILLUSTRATIONS

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PREFACE TO VOLUME I

THE term Modern Engines applies not only to engines of modern design, or invented in recent years, but also to all engines found to serve useful and efficient purposes at this date. Many old devices and designs have survived, and are worthy of a place when presented in their more modern developments.

The most highly developed machinery cannot in every case be employed by the engineer. In many instances rough and ready methods are necessary, and efficiency is sacrificed for the sake of convenience and simplicity. He has often to make use of what materials can be most readily found in his neighbourhood, and while a triple-expansion steam engine and a multitubular boiler would be highly appreciated and very economical as a prime mover, such an outfit would be in many cases impossible where less complicated and more primitive machinery would be quite successful. References are therefore made to older types still in practical use.

Each Volume of this work will contain some special feature. This First Volume discusses the fluid pressure machines operated by fluid impulse, by air, water, and steam flowing under a difference of pressure. These include that most important prime mover, the steam turbine, which therefore forms the special feature of Volume I. Being a new subject, and of great interests involved, the prior patents covering the fundamental inventions have been briefly noticed, as well as all the more recent inventions worthy of notice on this subject.

The application of the steam turbine to marine propulsion will be further discussed later on in this work under "Marine Engines."

The windmill is worthy of attention in connection with the possibility of transmitting and storing the energy generated. Combined with storage, the best site for the mill may be chosen, and the energy delivered where required at any reasonable distance.

The water turbines selected for description are the latest of their types, with special reference to governing. Injectors and centrifugal pumps are shortly discussed. These, although not prime movers, are important impulse machines closely connected with prime movers operating by momentum and fluid velocities.

With a description of the best rotary piston engines which have hitherto been made, this Volume concludes the survey of the more elementary prime movers and their accessories.

The next Volume will contain a full treatment of internal combustion engines—gas and oil, also of hot air and furnace gas engines; motor car engines and locomotive engines will follow.

RANKIN KENNEDY.

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MODERN ENGINES AND POWER GENERATORS

CHAPTER I

INTRODUCTORY

THE heat engine has been the theme of many great treatises. Like most other scientific subjects, it is easier to theorise and pursue mathematical investigation on assumed conditions than it is to invent, improve, design, and construct working engines or other practical appliances. While theories and mathematical conclusions are valuable guides, the results of actual machinery in practical tests are of immensely greater value, so that, while giving due weight to the theoretical side of the question, we shall devote this work principally to inventions, improvements, designs, and constructions in prime movers.

We know only two types of prime movers — the heat engine and the electric engine. Windmills and water wheels derive their energy primarily from the heat of the sun. Steam, oil, and gas engines derive their energy from fuel heat; they are all heat engines.

The electric engine has yet to be made practicable. The only direct source of electrical energy is the Voltaic battery, and that supplies only a small quantity at immense expense. All the electric motors so much in use are driven by heat engines; they are at present only transmittors of the power or energy of heat engines. By their aid, however, we are enabled to accomplish work with the heat engines which cannot be done by any other means. The simplest and earliest heat engines were driven by wind and water, and these have again been made more useful by the electric motor, transmitting their power to more convenient places where it can be better used.

Within recent times considerable advances have been made in prime movers; the dreams of early inventors have been realised in the steam turbine, the rotary steam engine, the gas engine, the oil engine, and motor car engines. Power from heat engines transmitted electrically to electric motors has also revolutionised the problems of rapid transit on street railways, underground railways, and is at present forcing the attention of main line railways.

The education of modern engineers also attracts much attention. By this is not merely meant the preparation of youths who intend to adopt engineering as a profession

or trade, but the education of the more important class—the owners, controllers, and employers in engineering works. The staff of a concern may be ever so well educated and skilled without much progress being made if the controlling head is not keenly alive to improvements and new inventions. There is need for more alacrity in adopting new and improved methods and machinery, which may be due to want of knowledge regarding them. There is no standing still in scientific manufactures at this date.

No employers or controllers can calculate, like their forefathers, upon making certain articles without improvements for a long lifetime; for good or ill those days are gone. To illustrate the meaning of these remarks two instances in the author's intimate knowledge may be quoted—first, the now highly successful electric tramways in our cities have been, 90 per cent. of them, equipped by American engineers with American machinery. Yet there is nothing in the equipment which can be said to have originated in America. The overhead trolley, the electromotor, the controller, the dynamo, and all the special electrical apparatus were well known to every electrician about twenty years ago in Great Britain, and it is quite safe to say that a score of skilled electrical engineers in Britain could have been found in the year 1888, any one of whom could have designed and laid out an electric tramway—dynamos, motors, trolleys, and all. There was no lack of education and skill among the youths nor among experts, but no employers or controllers with manufacturing facilities and capital could be found to take up what has been a very lucrative business to the Americans. The Americans have followed up their success by establishing large factories in England, so as to retain their hold upon the tramway and railway electric business.

Take also letterpress printing machinery: the largest newspapers, and some smaller ones, are all printed on American presses. A new printing establishment putting in new machinery nowadays must have the quickest and best working presses, machines with continuous revolving cylinders and two revolution or perfector machines. Three large press manufacturers in America have been shipping these improved presses in quantity to our printing establishments, simply because the home manufacturers go on making presses without improvements, and not at all for any lack of skill and knowledge in the younger generation of engineers.

Whatever may be the cause of it, there can be no doubt that the class to whom we must look for the encouragement of the advances in practical engineering require educating, in some way or another, to open their eyes to the value of promptly seizing upon every opportunity for securing improvements. It is very well for a country to have the honour and glory as the birthplace of improvements, but it is better, very much, if its captains of industry take advantage of them in time to advance the trade and industry of the country before outsiders step in.

It may be said that capitalists and captains of industry can avail themselves of expert advice; that, however, while the best course to take in ordinary matters, is not successful in questions regarding the adoption of new methods or manufactures. The expert desires to be, above all things, a "safe man," and naturally is conservative and timid when faced with responsibilities involved in new departures.

If every speculator consulted the family lawyer on the question of his investments there would be no enterprise in this world, no risks, no new departures. It is the robust, acute man, with a clear mind and depending upon his own education and knowledge, who sees and grasps the possibilities of a step in advance; and we want some powerful stimulant to arouse a class of capitalists and controllers in this country to bring out the best that is in the people for their own good, instead of supinely allowing foreigners to run away with all the inventions and improvements.

Fiscal arrangements are powerless in these matters: a bolstered-up, pampered industry makes no progress; it does not require to progress; nothing but energetic educated enterprising business can affect the situation. Writings have often proved of value in directing movements, especially in scientific work; to some extent this

work will present a résumé of up-to-date improvements, some already developed, others awaiting development. All this is somewhat of a digression; the excuse for it is that mechanical and electrical sciences are in their practical application retarded or accelerated largely by the enterprise of the capitalists. And it is hoped the treatment of the subject in these volumes may help to enlighten many who would bring up their knowledge to present date.

With the advent of the modern steam turbine the steam engine has arrived at its meridian; and the new Hult rotary engine may also give another finishing touch to the great edifice of improvements on steam engines. These two departures are worthy of the closest attention. But other agents besides steam are calling for recognition. The internal combustion engine in its many forms has reached a stage where it will also offer itself as a powerful competitor with steam.

Then the electrical transmission of power has brought the great water powers of the world down from their inaccessible fastnesses in the mountains to the workshops on the plains. It is no wild dream, but quite within the range of practical engineering, to say that we could transmit the energy of Norwegian waterfalls by submarine cables to Edinburgh to light the city and run its tramways. The utilisation of our own Highland lochs and waterfalls will come before that scheme.

A brief sketch of the history of the subject may precede the description of prime movers. It is proposed to describe the prime movers in their order first, and then to treat details afterwards. That the ancients became acquainted with means for moving huge masses to great heights and over long distances we have ample proof in the pyramids and ancient ruins. All primitive machinery was at first moved by animal and human power, using the machinery of levers, wedges, inclined planes, rollers, ropes, and other early inventions. With these engines man early discovered that if a weight of material could be moved at all, however small a distance in a day, he could move it any distance given plenty of time; and that, no doubt, is the explanation of the vast works carried out by such simple engines. Time was of no value; a few generations of men spent on a great work was taken then as a matter of course.

Wind power was no doubt first observed on boats upon the waters. The wind would move a boat without a sail, and this would naturally suggest a sail to increase the effect, and from a sail on a boat to a sail on a windmill of the old type is an easy transition. Water power was very early used to assist man in his efforts to move things. The Chinese records prove that water power wheels were invented by them long before the Christian era.

In ancient times intellectual contemplation and philosophical speculations were considered dignified and worthy of the most active and learned minds, while practical inventions and arts were considered as unworthy of notice in the ancient biographies and histories, and during the dark Middle Ages practical engineering and science were strangled by superstition and false religion. Galileo had to deny the motion of the earth round the sun to save his life. Owing to these prejudicial effects we cannot trace the earliest beginnings of prime movers; even the steam engine may have been invented long before any record we can trace.

The supposed higher dignity of mental contemplation and philosophical speculation still exists to some extent in ancient universities, but from the time of the sixteenth century the progress of mechanical invention, scientific discovery, and application to the work of the world has been very rapid and wonderful.

Heat engines are first found mentioned in Hero's *Pneumatics*, written 130 B.C., in which he describes a steam turbine of quite a practicable design, and also a steam pump for water raising. Another work, published in 1601, on pneumatics, by Battista della Porta, describes a steam water pump much the same as Hero's, but working with a vacuum to raise the water into a receiver, from which it was expelled by steam.

In 1629 Branca described and operated a steam turbine in which a steam jet drove

a wheel with vanes. He managed to explode his steam boiler, and was shut up as a prisoner on the plea that he must be mad. In England the Marquis of Worcester set up and worked a steam pump at Vauxhall in 1656. In 1697 Savery improved Worcester's pump, and it was introduced and much used for mine drainage. This seventeenth century, therefore, proved one in which the steam engine inventors were allowed to make known and use their engines; but so far these engines were steam turbines or water pumps, in which the steam pressure acted directly on the water and forced it up.

The next step in the evolution of the heat engine was one which was bound to come as better mechanical construction advanced. As soon as mechanics became capable of forming cylinders, rods, plates, and other pieces of metal with some accuracy of shape and form, it became possible to advance the heat engine. In 1690 Denis Papin had succeeded in constructing a cylinder and piston, the prototype of all our powerful steam engines up to within ten years ago.

He made a cylinder and fitted a piston into it, and in the lower part of the cylinder he placed some water; on placing the cylinder on a fire the steam produced forced up the piston, on swinging the cylinder from off the fire the steam condensed and the piston was forced down again. Here we have the elements of the reciprocating steam engine. And Papin, true to the inventor's instincts, enthusiastically describes his dreams of steam pumps, and, by rack and pinion, turning wheels and driving paddles on ships, all of which were quite feasible in his mind, but far beyond the skill of the mechanics of his day to put into practice.

It is to this day even a remarkable fact that inventors are sometimes ahead of the practical mechanic. Many an inventor conceives a valuable improvement, works it out in his own mind, and finds that the resources of the constructors and mechanics are not sufficient to realise his schemes.

All the early steam turbine inventors were baffled in their attempts, simply for want of tools and machinery sufficiently good to make their turbines. The recent success of the steam turbine has been more due to the refinement of modern machine tools and accuracy of metal-working machinery than to any new principles or inventions.

Nowadays the inventor has many advantages denied to the old pioneers,—machine tools are now made to produce almost anything in metal with the utmost precision, however complicated. A piston and cylinder are now fitted to the $\frac{1}{1000}$ of an inch easily, while James Watt was delighted when he succeeded in getting a fit so close that he could not slip a half-crown between the piston and cylinder.

Papin invented also a safety valve for his boiler, and so escaped the misfortunes of poor Branca.

Then followed Savery, Newcomen, and Cawley, who combined the separate boiler, cylinder, and piston with condensing water injected into the cylinder. Potter added the self-acting valves; Leupold and Smeaton added improvements, so that at the end of the seventeenth century the steam engine had become a well-known useful machine. Although frightfully inefficient, it had plenty of power, and did good work.

James Watt took up the question in 1759, and ten years after filed the most important patent specification the world ever saw, describing his improvements in steam engines. And after all, it was to Scotland and a Scotchman we are indebted for the final solution of the steam reciprocating engine problems; aided, however, by Matthew Boulton's financial assistance and personal energy, without which it must be admitted that James Watt's great genius would have been lost. Dr. Roebuck, of Carron Iron Works, had helped Watt at the outset, and enabled him to make a start. Boulton was a discoverer, and his greatest discoveries were James Watt, and later on Murdoch, the inventor of coal gas lighting. Matthew Boulton and James Watt carried out their great engineering work in Birmingham, and they had a long and bitter fight in the law courts to maintain their rights to Watt's inventions.

The chief claims of Watt's patent of 1769 are worth recording here. In Watt's own words—

"First, That vessel in which the powers of steam are to be employed to work the engines, which is called the cylinder in common fire engines, and which I call the steam vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first, by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

"Secondly, In engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam vessels or cylinders, although occasionally communicating with them; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water, or other cold bodies.

"Thirdly, Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam vessels or condensers by means of pumps wrought by the engines themselves, or otherwise.

"Fourthly, I intend, in many cases, to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner in which the pressure of the atmosphere is now employed in common fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the air after it has done its office.

"Lastly, Instead of using water to render the pistons and other parts of the engines air and steam tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state."

Watt afterwards proceeded to detail inventions,—the pendulum engine governor, the engine power indicator, the parallel motion, the expansive working by cut-off at a fraction of the piston stroke, the double acting engine, the butterfly valve, and other inventions, brought the engine to great perfection. It may seem strange that such a simple thing as a crank for converting reciprocating to rotary motion was unknown in Watt's time, yet it seems it was not known then, for it was invented by Watt, but pirated and patented by another.

When one considers that only a little more than one hundred years ago a crank and connecting rod and fly-wheel were new things, the extraordinary progress made between 1785 and 1885 can be realised.

It is of little interest to follow the history of the steam engine after Watt's time; all subsequent improvements have been mere details, and more due to taking advantage of improved materials and machine tools for better construction, using higher pressure, greater expansion, and multiple cylinders.

The reciprocating engine has reached its zenith in our day, and will likely hold the first place in many uses for many years to come, but the march of improvement never ceases. The steam turbine, so long impossible for want of means to properly construct it, has at last been evolved, and is now beginning to compete successfully with the Watt engine, which it is destined to supersede altogether in the steam-engine world. The rotary steam engine has been the dream of every engineer; even Watt spent much time upon it, but it has hitherto baffled all inventors. Recently, however, a new rotary engine has been constructed which in an exceedingly beautiful and simple manner overcomes the chief difficulty, that is, the internal friction of the revolving piston, so that hopes are raised also in this direction.

Steam, however, has its limitations, now well defined by scientists, and the point seems almost to be reached beyond which nature puts a stop to further improvement.

This is not to be wondered at, for during one hundred and fifty years the steam engine and its details have been the subject of tens of thousands of improvements and patents, and the research of physicists of the highest ability, mathematicians, mechanicians, and experts on steam.

Another type of heat engine was early proposed, and has been the subject of much study, namely, the "internal combustion engine." This title includes all that large class of engines in which the working fluid is hot air, heated in the cylinder by combustion with gas, or gasified oils, or fuel dust.

The fuel being burned in the cylinder direct, the efficiency should be higher than that of the steam engine, and it is so; but the high efficiency which might be expected has not yet been reached, a good deal of room for improvement still exists in these hot air internal combustion engines. High piston speed should be aimed at, and proper balancing and cushioning of the reciprocating parts should receive more attention.

The simple external combustion hot air engine has a small field of usefulness, but is of no account for large powers.

The designs of engines, steam, gas, oil, and hot air, are very numerous; adopted to many various purposes, we shall only study the modern types, leaving out the old beam, oscillating, grasshopper, side-lever, steeple, and other obsolete types, most of these designs being made to suit the tools of the builder. Thus if an engine maker had no large planing machine for iron slides, he designed the engines to be made without planed slides. Turning and boring tools preceded shaping and planing tools, hence old engines were designed with an eye to do all the machined work for them in the lathe, and all the motion gear in the forge or smithy.

The other source of energy, electricity, has become of great importance as a secondary power. Electricity cannot yet be liberated from fuel, like heat, by simple combustion. The Voltaic battery is the only available source of electricity direct from fuel, and as it uses expensive fuel with much waste it is not of commercial importance for power purposes. But by converting the power of a heat engine into electrical energy we obtain the most economical and ready means for transmitting energy for power and light. For this purpose it is only necessary to attach a dynamo-electric machine to be driven by the heat engine.

As a secondary power, electricity has a great field of usefulness. It also converts the heat engine power into a force which can be applied to chemical and metallurgical purposes in producing aluminium and refining copper. The production of caustic soda, potass, and carbide of lime, nitric acid, carborundum, potass chloride, and many other processes are capable of being carried out electrically.

As a secondary engine, the compressed air engine is also of much use, especially for drilling, riveting, and caulking in boiler and shipbuilding work. Also in rock-drilling in mines we shall have occasion to consider compressed air engines very fully.

In the case of the internal combustion engines, we have in the oil engine a complete power generator, in itself requiring no separate gas or other generator; hence its extremely suitable application to motor cars.

But gas engines require gas generators. These are of much interest to engineers, as offering a means of producing gas for power purposes from cheap fuel by a simple and cheap process.

The steam boiler is a necessary part of the steam plant, and has received almost as much attention as the engine; in its many forms adapted to many different purposes we have a large subject to study of very great interest. The fuels available for producing heat are not many,—coal, wood, oil, natural gas, waste products such as straw, cane, sawdust, and dust destructors. To coal the British Isles are indebted for their huge manufacturing industry; it has been cheap and plentiful, giving about 14,000 B. th. units per lb. burned. When coal becomes scarce, or is at last all used up in these islands, a vast change in the industrial world must have occurred. Coal is

being used up at an enormous rate, which goes on increasing yearly. The only hope is that engineers will soon discover an engine in which coal can be more economically used; the best heat engines are at present very wasteful of fuel, the steam engine wasting about 80 per cent. of its fuel at this present day. What a field for invention and improvement this fact discloses!

In many manufacturing processes fuel is also recklessly wasted to a large extent. In the iron-blast furnaces coal consumption has been gradually reduced, and the utilisation of the furnace gases for power purposes promises a further saving. Coke is used for many purposes, and, as at present made, all the gaseous and liquid products are wasted in its manufacture; the blazing coke heaps and coke ovens are a terrible waste of fuel. The prodigal abuse of Nature's greatest store of energy on earth is pitiful. Coal can never be restored in the ages of man, and a coalless country can never be anything but a thinly populated agricultural and pastoral land. The saving of coal is one of the engineer's chiefest problems. Mineral oil has been found plentiful in some localities in Russia and America, and can be made from Scottish shale. As a fuel, it is far inferior to coal, but it is convenient for transhipment. Whether the oil supplies will last for long or short times we cannot know. Texas seems to have a large natural supply of crude petroleum, which, being of little use for anything else, is sold cheap for power purposes. In a suitable engine it gives 1 horse-power-hour for one farthing.

In colonial countries boilers are in many cases fired by straw and wood, the furnaces being constructed specially for the purpose, as the fuel is very bulky compared with its output of heat. At timber works, saw mills, etc. the waste wood and sawdust is used for fuel to work the engines.

The refuse destructor in large towns and cities produces heat sufficient for raising a large quantity of steam, which in some cases is used for electric lighting purposes.

In tropical countries direct sun heat has been used for raising steam for small engines, the boiler being placed in the focus of a large mirror.

All these fuels and their uses are well worthy of the engineer's attention; and the accumulated knowledge of their values, and the best uses to make of them, is to be gathered in this work in later chapters.

Condensers, economisers, pumps, valves, and all the many accessories of modern prime movers we will discuss in their proper places.

CHAPTER II

PRIME MOVERS, CLASSIFICATION OF

In treating this extensive subject there are two methods of procedure. First, to begin with the sources of energy—fuel, heat, electricity, and the elements of mechanism, thermodynamics, electrodynamics, fluid pressures; and thence to engine designs and constructions. This course, however suitable as an educational curriculum, is not of so much practical value as the second method; in this we follow more closely the natural course of development of the prime mover. Engines were successfully constructed and used without a knowledge of the exact sciences connected with their theory of action, this knowledge forerunning the theories. As an analogy, we may refer to the science of physiography. A surveyor could possibly by scientific measurements and observations and exact calculations delineate on a map the approximate course of a river from his knowledge of the watershed; while an ordinary traveller, beginning at the mouth of the river, could by simple observations obtain a far more accurate and exact knowledge by following the course from mouth to source. In the same way we shall begin where the pioneers in engine making started, namely, where they commenced to make engines without inquiring closely into the more abstruse problems. These deeper scientific questions follow naturally as the subject develops. The fluid pressure engines are primarily engines the working substance in which are fluids.

CLASSES OF PRIME MOVERS

Fluids are of two kinds—liquids and gases.

1. *Natural Fluid Pressure Engines*

Natural fluid pressures are used in windmills—air pressure; and in water engines, wheels, and turbines—liquid pressure.

2. *Steam Engines*

Artificial fluid pressure engines are those in which the pressure is produced by heat from fuel combustion, and comprise steam and hot air engines. Steam engines are of three classes:—

(1) Steam turbines.
(2) Reciprocating piston engines.
(3) Rotary piston engines.

3. *Hot Air Engines*

Hot air engines are of two kinds :—

(1) External combustion engines, in which the fuel is burned outside the engine.

(2) Internal combustion engines, in which the fuel is burned inside the engine ; to this class belongs the gas and oil engines.

4. *Electric Engines*

In this class the energy is applied by electric pressure generated from fuel by chemical combination, which may be properly described as combustion without heat, electric pressure being the result, instead of high temperature.

5. *Secondary Engines*

These are engines operated by fluid pressure or electric pressure, generated by some prime mover of the types belonging to Classes 1 and 2, such as compressed air, turbines driven by water which has previously been pumped up to a height by a windmill, water turbine, steam or gas engine, and electromotors driven from dynamos.

WINDMILLS

The windmill and water wheel naturally preceded the heat engine in practice. We therefore begin with these engines, and will follow in the order here given, with the consideration of the construction of the others.

Naturally, wind power would be the earliest, and windmills the first prime movers. The force of the wind is self-evident, and the first attempts to utilise it were no doubt in the direction of propelling boats by sails. At the present day sails and masts are still used on large vessels for propulsion. The sailing vessel can be moved in opposite directions by the same wind, and can sail on a course at an angle in part against the wind. This feat would at first be found by experiment ; but it can be demonstrated by the principle of the resolution of forces as given in text-books on Mechanics.

The old windmills so conspicuous on the landscape were mechanically simply four masts, with one sail on each, set on a rotating shaft. In a 10-mile breeze a 15-foot wheel gave about 1 horse-power, and the power given was roughly about proportional to the square of the diameter from tip to tip of the sails, and the best results obtained when the speed in feet of the sail tips was about 2.5 times the speed of the wind.

The old windmill did good work in its day, and its more scientific successor is capable of still better performances. Far from regarding the windmill as a sailing body like a boat, the modern engineer now considers it in its true light as a fluid pressure turbine. Like water power, the best sites for the wheel are generally far away from the points where the power is required, so that some system for transmitting the power must be adopted to obtain the best results. And again, as with all natural forces beyond the control of man, some system of storage must be employed if the force is to be ready for use daily at command.

In electricity we have both requirements provided—a means of storage and a means for transmission—so that we can store up the power when the wind blows ; and in designing the plant, place the wheel on the best available site, even miles from where the power is wanted, and then transmit the power by electric current from the wheel to the work to be done. This electrical transmission puts both wind and water power on quite a different footing from that on which they stood twenty years ago. At that time steam had carried all before it, but since then it has been found that

steam does not accomplish everything desired. It has its limitations, which are now clearly defined; and more attention is paid to other working fluids for prime movers.

Now, when we can choose the highest and most suitable site for a windmill, and transmit the power any reasonable distance for use, it becomes a more generally applicable prime mover.

Windmills are, however, very limited in power. A pleasant breeze flows about 10 miles per hour; and as that can be counted upon for about 8 hours a day on an average, it is taken as a minimum standard. The power, however, increases nearly as the square of the speed of the wind, so that the power varies largely.

In practice it is found that well-made modern mills and water pumps will do the work as under.

TABLE I.

Diameter of Sail in Feet.	Water raised in Gallons 100 Feet high per Hour with 10-mile Wind.	Horse-power in Work done.	Wind at 20 Miles per Hour Horse-power.
7	80	$\frac{1}{25}$th	$\frac{4}{25}$ths
10	200	$\frac{1}{10}$th	$\frac{2}{5}$ths
12	350	$\frac{1}{6}$th	$\frac{1}{3}$ths
16	700	$\frac{1}{3}$rd	$1\frac{1}{3}$rd
20	1200	$\frac{3}{4}$ths	3

The above figures are for mills driving water pumps direct without gearing.

Geared mills work the pumps slower, and are generally used where power is taken direct from the mill.

TABLE II.

Diameter of Sail in Feet.	Water raised 100 Feet per Hour 10-mile Wind.	Horse-power approximate with 20-mile Wind.	
	Gallons.		
25	2000	4	Actual Brake Horse-power.
30	3000	$5\frac{1}{2}$	
35	4500	$7\frac{1}{2}$	
40	6000	12	

The power should be as the square of the diameter of the wheel and as the square of the velocity of the wind. The horse-power in water raised in feet per hour equals $\frac{\text{gallons} \times 10 \times \text{feet height}}{60 \times 33{,}000}$, multiplying gallons by 10 to get lbs. Thus for a 25-foot mill $\frac{2000 \times 10 \times 100}{60 \times 33{,}000} = 1$ horse-power; and as the power is as the square of the velocity of wind, we would get for a 20-mile wind $\frac{20^2}{10^2} = 4$ horse-power at that velocity.

USEFUL FORMULA

The force of wind increases as the square of its velocity.

a = area exposed at right angles to the wind in square feet; F = force of the wind in lbs.; H = horse-power; and v = velocity of the plane a in direction of the wind, + when it moves opposite, and − when it moves with the wind.

$$F = 0.002288 a V^2, \qquad \text{when } v = o$$

$$F = 0.002288 a (V \pm v)^2 \qquad H = \frac{a v (V \pm v)^2}{240384.6}.$$

Example.—A rail train running E.N.E. 25 miles per hour exposes a surface of 1000 square feet to a pleasant brisk gale N.E. by N. Required the resistance to the train in the direction, it moves and the horse-power lost.

E.N.E. – N.E. by N. = 3 points = 33° 45′; V = 14 feet per second, a brisk gale; $v = 25 \times 1.467 = 36.6$ feet per second; and $F = 0.002288$ sin. 33° 45′ × 1000 $(14 + \cos. 33° 45' \times 36.6)^2 = 305.1$ lbs.

$$H = \frac{305.1 \times 36.6}{550} = 20 \text{ horses.}$$

The motions and effects of gases by the force of gravity are analogous to those of liquids.

The altitude or head of the atmosphere at uniform density will be the altitude of a column of water 33.95 feet divided by the specific gravity of the air, 0.0012046, or

$$\frac{33.95}{0.0012046} = 28,183 \text{ feet.}$$

The velocity due to this head will be $V = 8.02\sqrt{28,183} = 1346.4$ feet per second, the velocity with which the air will pass into a vacuum.

VELOCITY OF WIND

When air passes into an air of less density, the velocity of its passage is measured by the difference of their density.

H and h = density of the air in inches of mercury; t = temperature at the time of passage; and V = velocity of the wind in feet per second.

$$V = 1346.4\sqrt{\frac{H - h}{h}(1 + 0.00208t)}.$$

As the wind averages between 10 and 20 miles per hour, we might count on an average between 1 and 4 horse-power as the output of a mill of 25 feet diameter on a 30-foot tower placed in a favourable situation. These observations refer to the ordinary modern mill as shown in Fig. 1 ungeared, and in Fig. 2 geared, in which the sails are moving in a plane at right angles to the wind, and are set at an angle such that the tips of the sails move about 2.5 times the velocity of the wind. Thus the 25-foot wheel would in a 20-mile wind make 24 revolutions per minute, not an excessive speed. This type of wheel has the advantage over others of presenting every blade constantly to the cylinder of wind which it intercepts, and runs at a fairly high speed. The whole of the blade is not equally effective, however, as its effect decreases towards the centre, where the wind has less leverage, and in high gales it is apt to be damaged, so that attempts have been made to design other forms, which we will also describe.

In this present form the blades are flat metal sheets, geared so that they can be feathered in order to present them at different angles to the wind, in some cases to turn them edge on in high gales so as to reduce the pressure. This geared mill is shown in Fig. 2. The little windmill at the back winds the large wheel round, so that it always automatically presents the same face to the wind. The blades also automatically adjust themselves in heavy winds so as to prevent excessive speeds. These automatic gears are necessary on all large mill wheels of this type. The small wheel is known as the fantail. In some old mills it was little else than a tail, a flat board on a tail rod on the axis of the wheel.

Referring to Fig. 1, which illustrates the smaller mills ungeared for pumping water, made by Messrs. Robert Warner & Co., the mills are provided with fixed galvanised vanes, fixed to circular rings and arranged to turn partly out of the wind when the velocity ot the latter is higher than is required for the proper working of the mill. Fantails are provided to each mill for maintaining the sail to the wind. Suitable gearing is arranged for conveniently putting the mill into or out of work from the

ground level, so that the attendant need not go to the top except for the purposes of lubricating, which is very seldom required. The towers are made square (excepting for the larger sizes, which would be hexagon), having four corner posts of angle section, with suitable strong strengthening rings and bracings. These are made mainly of steel, wrought iron being used for bolts and other smaller parts. The towers are

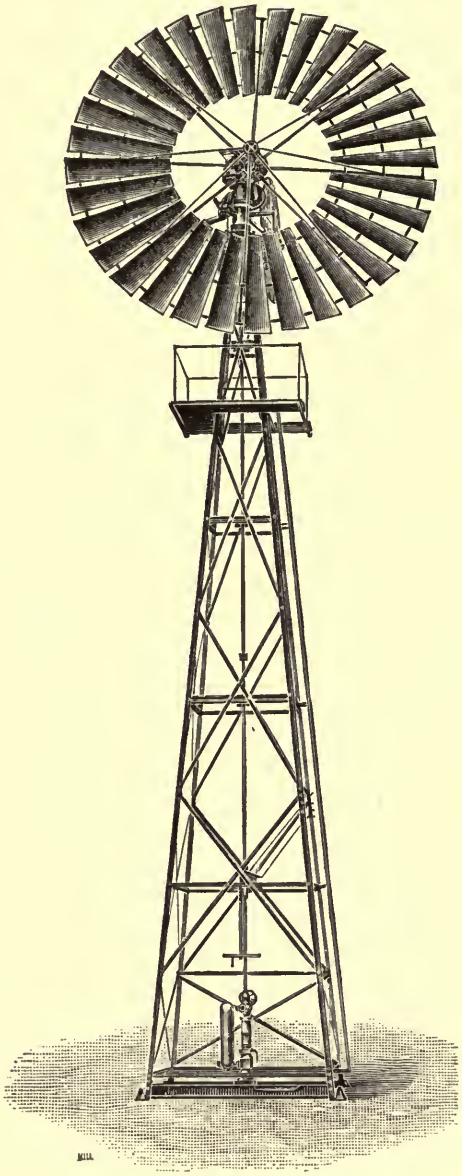


FIG. 1—Simple Ungeared Windmill.

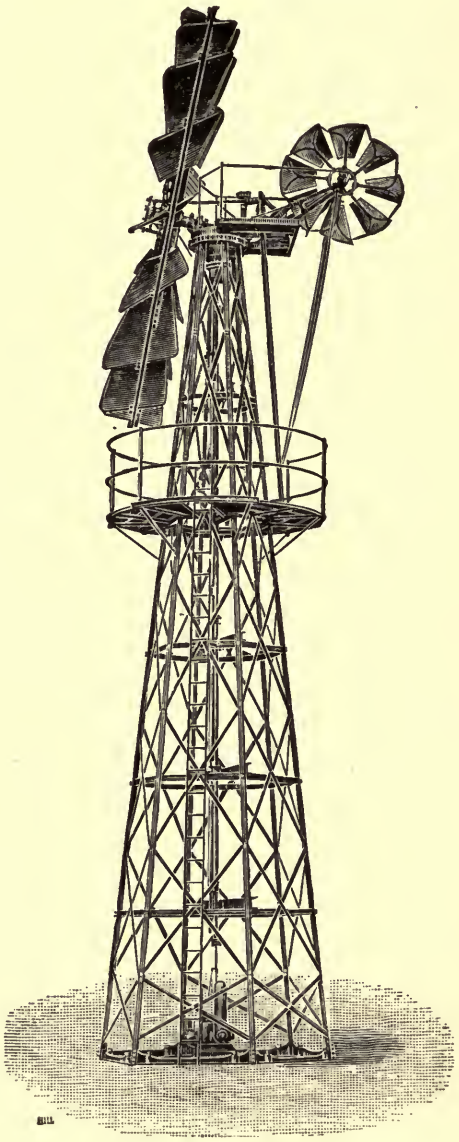


FIG. 2—Geared Windmill.

arranged for bolting down to four blocks, which may be of concrete, brickwork, or other suitable materials that may be available.

The sail and pump shafts are of best steel, the bearings, including the turntable and millhead, being very carefully designed so as to give the smallest amount of friction. Special attention is given to the parts requiring lubrication, so as to need the least possible attention.

The geared mills are constructed of larger sizes, from 20 to 40 feet in diameter. The Figure 2 is from a photograph of a 40-foot mill supplied to the Government for Egypt.

These mills are substantially built with very large wearing surfaces, which are well provided for means of lubrication, thus rendering them very durable. They are automatic in their action during heavy winds, so that excessive speeds which might cause damage are avoided. The winding gear consists of a geared back fantail which is far the most efficient and satisfactory type to adopt with geared windmills. They can also be quickly started and stopped by hand from ground level. The power is transmitted from the sail shaft to the ground or any other level of the tower by a pair of bevel wheels and upright shaft. The power can be taken from the latter or from a horizontal as may be most convenient. The towers are made mainly of steel, wrought iron being used for bolts and some of the smaller parts. The smaller towers are square and the larger ones hexagon, all having suitable strengthening rings and bracings, and are arranged for being bolted down to suitable foundations, such as concrete, brickwork, etc.

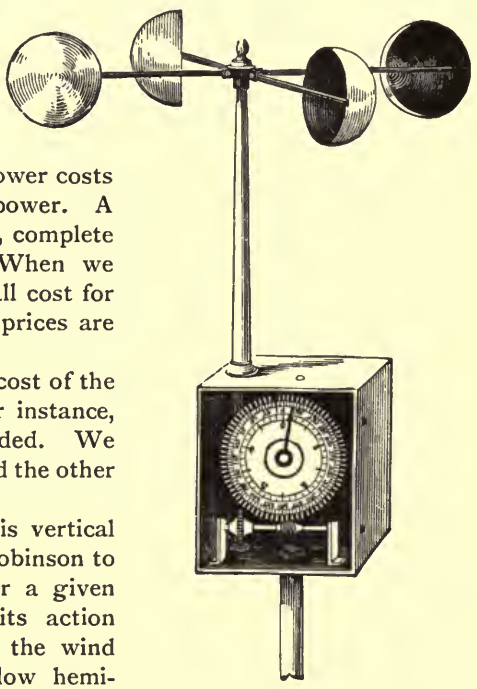


FIG. 3.—Robinson Anemometer.

The first cost of windmills is not high. A 20-foot mill ungeared with a 30-foot iron tower costs about £140, giving from about 1 to 3 horse-power. A 40-foot mill giving from 6 to 12 horse-power, complete with a 40-foot tower, costs about £470. When we consider the simplicity of the machine, its small cost for upkeep, and no cost whatever for fuel, these prices are not excessive.

These prices, however, do not include the cost of the power transmitting or utilising gear; say, for instance, the pump, if it is for water raising, is not included. We will return to this point after we have discussed the other two forms of wind wheels.

A wind wheel which rotates with its axis vertical and the wheel horizontal was designed by Robinson to register the average speed of the wind over a given period. This form of wheel depends for its action upon the difference of resistance offered to the wind by the concave and convex sides of a hollow hemisphere. The convex sides offer much less resistance, hence the wheel revolves in the direction pointed by the convex side. It is thus evident that the whole pressure of the wind is not available, as the back pressure on the convex side must be deducted from the forward pressure on the concave side; and the pressures act only for about one quarter of a revolution.

On the other hand, the speed can never exceed that of the wind, and the arms may be made long and strong. A considerable leverage can be given to the acting cup. The Robinson anemometer is shown in Fig. 3, as made by Casella.

On this same principle the wind wheel of Professor Blyth is designed. It is shown in Fig. 4, the hemispherical cups being replaced by wooden buckets of large size; the construction explains itself pretty clearly. The difficulty with these wheels is their great size required for comparatively small powers. A better design of the same type has been made in the form of a water wheel, a portion of which is shown in Fig. 5, giving the delineation of the vanes; and in Fig. 6, where it is seen mounted on a vertical axis on a tower. Here again the torque is due to the difference of resistance between the concave and convex sides of the vanes; only, the

one partially screens the other, so that the difference is greater than it is with cups Neither of these types are as efficient as these shown in Figs. 1 and 2.

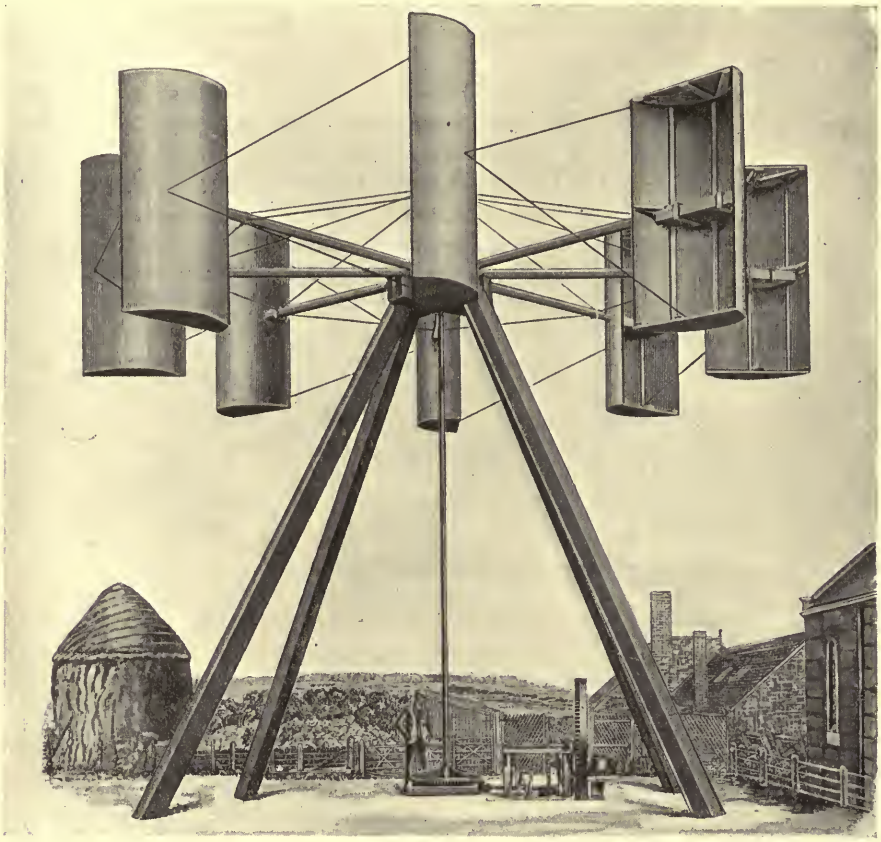


FIG. 4.—Blyth's Windmill.

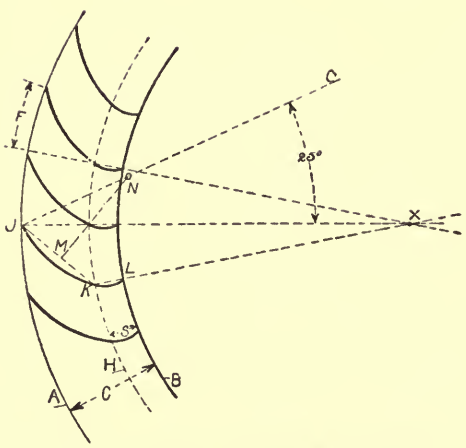


FIG. 5.—Diagram of Wind Wheel Vanes.

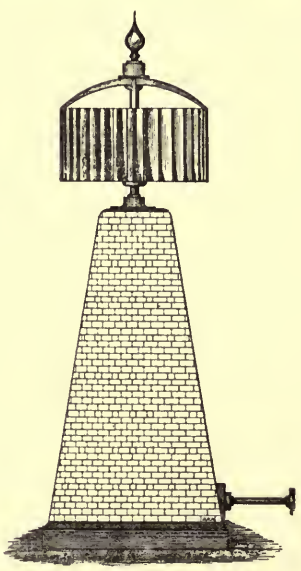


FIG. 6.—Horizontal Wind Wheels.

POWER TRANSMISSION GEAR FOR WIND WHEELS

The gear from the wheel shaft itself consists of a vertical shaft and bevel wheels, a pump and crank to work the plunger where simple water raising is required. As a rule, however, the water lies low, in a well or pond or river, while the best place for the wheel is high up. And as we cannot bring the water up, we must take the wheel down to the water, and thus it becomes less effective. This difficulty might be overcome by transmitting the power by some means from the high wind wheel down to the low water pump. Ropes and pulleys and compressed air have been used for this purpose, but they are expensive and inefficient, and the source of much trouble; of the two, compressed air is the best. In this system the wind wheel pumps air under pressure into a receiver, generally a cylindrical steel boiler. From this an underground iron pipe connects down to the pump placed at the water source, the compressed air here, by means of an air motor and water pipe combined (to be described under pumps), pumps the water up to a reservoir to the desired level, from whence it is supplied by gravitation to the users. For pumping purposes this is a satisfactory system, for both wind wheel and water pump can be placed just where they are most advantageous, and even more than one pump may be used from same wind wheel if several sources of water can be tapped.

For power purposes the same arrangement can also be used with advantage, and by providing large storage vessels for the air under pressure a fairly constant supply of air can be relied upon for daily work. This system for transmitting and utilising wind power possesses much to recommend it in Egypt, Westralia, and other places where fuel and water are scarce. The other system is the electric transmission system, in which the wind wheel drives a dynamo-electric generator, from which wires are run to the motor or motors where the power is required.

Like compressed air, it requires for its reliable working a storage reservoir, in the shape of an electric storage battery. And as the speed of the wheel varies with the power and velocity of the wind, a special battery must be used capable of charging between very great differences of electric pressures, so as to make use of all winds blowing between 10 and 30 miles an hour.

The problem to solve in the storage of the energy of the wind wheel is much the same for compressed air and electricity. In the case of compressed air the pump must work between great differences of speed; and as the torque required to move its pistons, rods, etc. must be sufficient to work it at a 10-mile breeze speed, it is evident—this torque being constant at all speeds—the speed of the wheel will go up nearly as the square of the wind velocity, and the increased work of the pump will be due to and proportional to the increased speed and not to increased torque; the variation in speed would be very large. A variable stroke compression pump would greatly reduce this large variation, but that is a difficulty in itself. The next best solution of the problem is to use a compound air pump with three or four cylinders and pistons, in which one, two, three, or four pumps may be worked from the wind wheel,—one on the lightest breeze, 10 miles, two on the 15-mile speed, three on the 20-mile, and four on the 25-mile speed,—each pump being also larger in capacity than the preceding one. By this means the speed may be very much moderated, for the torque required increases as pumps are put on, so that the work done is proportional to the wind pressure or torque.

The electrical dynamo generator for storing electricity into the battery acts to some extent (on a wind wheel) similarly to the compound pump. If the speed of the dynamo at the lowest speed of the wind wheel is such as to generate an electric pressure slightly greater than the battery pressure, then as the speed increases the torque required also increases, for the pressure of the generator and the current also increases into the battery. Hence the speed of the wind wheel is opposed by

this increasing drag put on by the generator, and the variation in speed much moderated. For the purpose the dynamo should be compound wound and with field magnets far below saturation-point at the small load. The battery must be calculated to take the charging current at the maximum speed without excessive heating.

The electrical points will be elucidated under their own headings.

If compressed air is employed for transmitting the power, then some further considerations must be taken into account. When air or any gas is compressed by force, sensible heat appears; the gas is raised in temperature during the compression. The question as to whence cometh the heat is an important one, which is discussed under the science of heat. In the meantime, suffice it to say that if the air is carried away in pipes or stored in a reservoir this heat is dissipated and lost by conduction and radiation, and the condensed gas in losing the heat falls in pressure to some extent. And further, we shall find that if the gas is afterwards expanded doing work in an engine, we must make good this loss of heat if we are to get efficiency out of the use of the compressed air; for the gas on expanding must get back the heat produced when it was compressed, and if this amount of heat is not supplied to it while expanding from some extraneous source, it takes the heat from itself and the containing pipes and cylinders and falls in temperature, so much that in air motors of any great power the passages, valves, and cylinders become filled with ice and snow, produced from the moisture in the air freezing from the extraction of the heat by the expanding air.

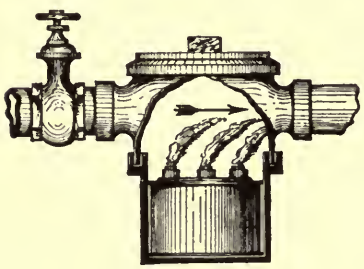


Fig. 7.—Reheater.

If compressed air is therefore allowed to cool down before use in an engine, then to get good efficiency we must provide some source of heat to "reheat" it before it enters the engine.

In fact, this absorption of heat by previously compressed gas which has been cooled under compression is the fundamental principle upon which some ice making and refrigeration by machinery is based. Air is compressed in these machines, then cooled as much as possible, then expanded, when it thereupon falls below zero temperature. This freezing air is then led through vessels, in which it abstracts the heat and produces ice.

In the air transmission of power not only is the ice and snow a nuisance in the engine, but it reduces the efficiency, for the mean pressure is less as the temperature falls. Reheating is therefore an important engineering question in compressed air power, which cannot be fully gone into in this chapter, but we may refer to a simple means for supplying the necessary heat suitable for simple transmission under the considerations of wind power.

Into the supply pipe next to the engine a steel box is inserted large enough to hold an oil lamp with several large wicks. The box has a movable cover, easily secured, air tight, and capable of withstanding the maximum air pressure; under these conditions all the air entering the engine must pass over the lamp wicks. To start the engine the lamp is ignited, the cover shut down, and the air turned on; the engine starts, and as the air passes over the lamp it maintains the combustion, and at the same time heats the air entering the engine. This is an example of an internal combustion heater in which the products of combustion also pass into the engine. There is no chimney loss, hence it is very efficient.

The efficiency of the air motor is much improved, and no troubles from ice and snow occur. Diagram Fig. 7 shows the apparatus.

Wherever wind power is seriously considered it is worth while weighing the advantages of the two systems of storage, compressed air and electric. In out of the way

places compressed air seems the simpler system to maintain, as it is wholly mechanical in operation and of common mechanism in construction; while the electric system is the more flexible in application, overhead wires on poles carrying the power wherever required.

CONSTRUCTION OF WIND WHEELS

The old wheels had only four or six sails carried on poles like masts. The sails were canvas spread on a brander of wood, with cords and pulleys whereby the sails could be "reefed" to reduce the pressure in a heavy gale. Wood was the material used throughout. They are to be seen yet in the eastern counties of England, and several good examples are still working in the Orkney Islands grinding grain.

The modern wheels herein illustrated are built of mild steel, strong, compact, and of light weight, and the principles of design are the same as those of a water turbine, for they are fluid pressure turbines of a simple design.

Properly designed, the blades of the wheel should be made of sheet steel, and curved so as to gradually change the direction of the air stream striking them. For common practical work, and owing to the facility with which they can be feathered, flat blades are used, as shown in Figs. 1 and 2. Curved blades cannot be turned edge on to the wind, so that in a heavy gale they would be damaged; but in any wind wheel of importance arrangements can be made whereby the whole wheel can be turned edge on to the blast.

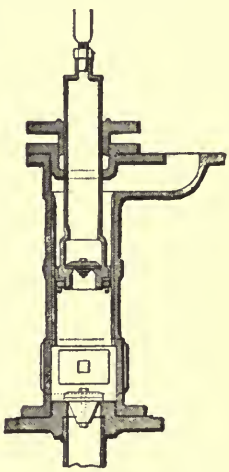


FIG. 8.—Windmill Pump.

The blades when curved should be shaped like the blades of a parallel flow water wheel, as shown in Fig. 5. The pump to be employed for compression of air in cases where that system of power storage is adopted must be simple and effective. The whole question of compressed air power and apparatus will be fully treated later; meanwhile, it will suffice to show a very simple form of water pump suitable for the windmill (Fig. 8). The pump and the driving shaft are fixed conveniently to a cast-iron hollow column, which forms a receiver of small capacity. It is a simple trunk piston pump with large valves. For large storage of air, wrought-iron drums are of greatest capacity, with a minimum weight; those should be made of a size easily transported, and a number connected together by a pipe to obtain the desired capacity.

For electric storage the dynamo chosen should be a series wound machine, feeding into a battery of large capacity, and the terminal electric power delivery at the battery should be at least 2 volts less than the dynamo terminal power delivery when the mill is running on the lowest speed it may be designed for, and an automatic cut-out must be in circuit in order to open the circuit the moment the battery and dynamo power deliveries are equal. By this arrangement, as the speed increases with the speed of the wind, the torque will increase also.

In charging, the power delivery of each cell may be 2.5 volts, and may be 2.7 volts; in fact, the charging volts may range from 2.2 to 2.8 volts. At 2.2 volts the current entering the cells will be very small, while at 2.8 volts it may be very large, the difference of 0.6 volt giving a large current through the cell's very small resistance. The cell may have a resistance of 0.01 ohm, so that the current would be $\frac{0.6}{0.01} = 60$ amperes.

Any series wound dynamo will not do for this work. Common series dynamos are,

as a rule, made with large armatures and small fields, so that they are nearly constant current generators. A dynamo for a windmill must have, on the contrary, a large field and small armature, so that the field strength goes on increasing proportionally with increased speed. Such machines are not to be found in stock, but must be specially designed for the wind wheel to be used with it.

Attempts have been made to combine two machines to be coupled in series parallel for different speeds on a windmill to be carried by ships, but the simple series dynamo specially designed is much better and far simpler.

The electric system of storage is the best for wind power, and is well worthy of the consideration of engineers; while the compressed air system offers advantages in colonial and outlying work where the electric battery and motors might not get the required skilled attention.

In the spring of this year the Royal Agricultural Society of Great Britain took up the question of wind engines. The following particulars and illustrations are from their report thereon:—

Having regard to the increasing demand for an inexpensive pumping plant, and the very material alteration in the design of the modern wind engine, the society decided to institute trials of wind pumping engines in competition for prizes, so that trustworthy data might be provided for the guidance of intending users. No independent competitive trials of these engines have previously been carried out in this country.

As the use to which these engines are principally applied is that of pumping, and as with an engine running for several hours throughout the week a very considerable amount of water may be pumped with comparatively small power, the limit of 4 horse-power was decided upon as a maximum. It will be seen from the results of the trials that this was a very ample allowance, as none of the competing mills attained such power under the stipulated wind velocity, which was fixed at ten miles per hour. This velocity is considerably in excess of the mean velocity at inland stations throughout the year in this country. On the other hand, there are periods at which the engines should be able to work efficiently at very much higher velocities. It was obviously undesirable to fix upon a high wind velocity, as it is all important that, though the engine should control itself and work efficiently in a high wind, it should also run in light winds, which are generally prevalent during the driest periods of the year when there is the greatest necessity for pumping.

With a view to ensure that the trials should be held under the most variable conditions as to wind velocity, they were fixed to take place during March and April. The records of the wind velocities during the period of the trials proved the wisdom of this choice of season, as during the greater part of the time there was plenty of wind, with just a few days of light wind, which afforded the opportunity that was desired of comparison between the several engines under those conditions.

To gauge the power developed by each engine, it was stipulated that the amount of water pumped against a head of 200 feet should be measured, and that such should be taken as the standard upon which to compare the work done by each.

It is not to be assumed that 200 feet head is in any way the average head at which such engines would ordinarily work. In most cases they would raise the water to a very much smaller height. On the other hand, they might be required to raise water to a greater height. This is a point which merely affects the size of the pump; and the particular head of 200 feet was adopted in order to keep the size of the pump and its connections within moderate dimensions; 11 gallons per minute representing a horse-power.

Twenty-two competitors entered, and the tests came off as arranged on the society's permanent show ground at Ealing. Sixteen of the competitors for various reasons dropped out of the contest before the final, and six entered for the final tests, with the result shown in Table III.

TABLE III.—PERFORMANCE OF WIND ENGINES IN PUMPING WATER.

		I. No. 3. Goold, Shapley, & Muir Co.	II. No. 7. Messrs. Thomas & Son.	No. 8. Mr. John W. Titt.	No. 14. Messrs. R. Warner & Co.	No. 16. Mr. John W. Titt.	No. 17. Messrs. Henry Sykes Ltd.
Price		£70	£77	£61, 7s. 6d.	£79	£74, 10s.	£106
Diameter of Wheel .		16 feet.	16 feet.	16 feet.	16 feet.	16 feet.	16 feet.
1903.	Wind Velocity.	Gallons.	Gallons.	Gallons.	Gallons.	Gallons.	Gallons.
March 4 .	15	...	3137	...	3563	...	...
,, 5 .	5	...	1030	...	343	...	...
,, 6 .	10	...	2752	...	3950	...	...
,, 7 .	12	...	3019	...	4207	...	...
,, 9 .	12	...	...	...	...	...	3151
,, 10 .	8	...	...	2214	...	...	2045
,, 11 .	6	1,428	...	1181	...	...	895
,, 12 .	8	2,545	...	1913	...	...	1733
,, 13 .	9	3,608	...	2309	...	...	2268
,, 14 .	3	...	...	...	. .	15	...
,, 16 .	15	...	...	...	...	3772	...
,, 17 .	22	...	...	...	...	3339	...
,, 18 .	20	...	2094	...	3140	4135	...
,, 19 .	20	...	4746	...	5653	4203	...
,, 20 .	18	...	4520	...	5250	...	...
,, 21 .	18	...	3912	...	4874	...	...
,, 23 .	22	...	4525	...	3640	...	5244
,, 24 .	12	...	...	...	...	...	2781
,, 25 .	15	...	...	...	...	...	4210
,, 26 .	25	...	...	...	...	...	5445
,, 27 .	8	2,891	...	...	...	1248	...
,, 28 .	24	10,047	...	4447	...	3470	...
,, 30 .	22	7,632	..	4399	...	3361	...
,, 31 .	7	3,160	...	1850	...	1373	...

The Performances of the Engines

In accordance with the regulations the following specific points were taken into account:—

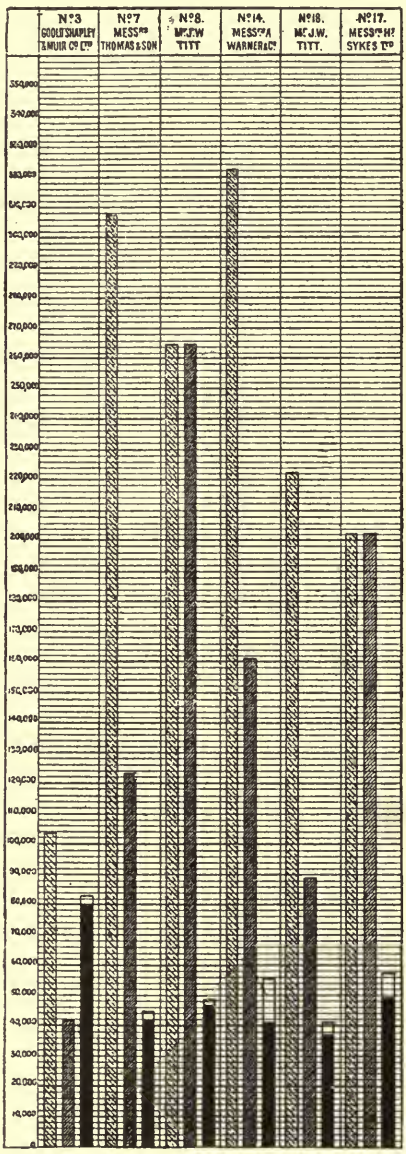
1. The design and stability of the tower.
2. The cost of the foundations and the price of the engine and pump.
3. The regulation of the working and control of the machinery.
4. The automatic governing of the engine in varying winds.
5. The efficiency as represented by the amount of water delivered under the specified conditions.
6. The cost of maintenance in respect of repairs incidental to ordinary wear.
7. The ease of erection.

The estimation of the merits of the several engines in respect of these seven points depends partly upon the examination of the designs and partly upon experience of the working of the engines during the trials.

It may here be noted that the performance was judged upon the working of the engines and pumps as combined machines. The selection of a pump inappropriate for

the particular engine under the condition of the trials placed the competitor in a most disadvantageous position.

Two of the machines, Nos. 14 and 17, were provided with arrangements for adjusting the length of stroke of the pump while running. For the purposes of the competition the strokes of these pumps were fixed, at the outset, at the length estimated by the exhibitors to give the best results under the condition of the trials, and were not



EXPLANATION.

= Number of revolutions of wheel.
= Number of strokes of pump.
= Quantity of water in galls.
= Slip.

PARTICULARS OF PUMPS.

No. 3 =	4 inches by	22 inches.		Double-acting.
,, 7 =	4 ,,	8	,,	Single-acting.
,, 8 =	$3\frac{1}{4}$,,	$6\frac{1}{8}$	,,	,,
,, 14 =	$3\frac{1}{2}$,,	5	,,	Double-acting.
,, 16 =	$4\frac{1}{2}$,,	8	,,	Single-acting.
,, 17 =	$2\frac{1}{2}$,,	8	,,	Double-acting.

NOTE.—Nos. 8 and 16 are "Bucket and Plunger" Pumps.

FULL BORE CAPACITY PER STROKE.

No. 3	=	2	gallons.
,, 7	=	.3636	,,
,, 8	=	.183	,,
,, 14	=	.348	,,
,, 16	=	.4601	,,
,, 17	=	.284	,,

RATIO BETWEEN WHEEL AND PUMP.

No. 3	=	$2\frac{1}{2}$ to 1.
,, 7	=	$2\frac{1}{2}$,, 1.
,, 8	=	1 ,, 1.
,, 14	=	2 ,, 1.
,, 16	=	$2\frac{1}{2}$,, 1.
,, 17	=	1 ,, 1.

FIG. 9.—Diagram Illustrating the Working of six selected Wind Engines during Final Trials. 150 working hours.

subsequently adjusted. Credit for this facility of adjustment was given in estimating the design. It would not be fully represented in the recorded performance.

Others of the exhibits had arrangements in the head gear for adjusting the length of stroke, as will be seen from the descriptions given above.

One requirement was regarded as inexorable. The engine and pump were expected to stand the strain of the two months' run with only such attention as might be supposed to be available on the spot in ordinary conditions of working. Any breakdown that required the maker to be called in to get the machine into proper working order again was regarded as terminating the competition so far as that machine was concerned, on the ground of inadequate stability and durability.

The final stage of the competition dealt with the selection of two mills for the First and Second Prizes respectively from among the selected six. The diagram (Fig. 9) represents the results obtained by the second month's run of the six engines. It shows the number of revolutions made by each of the engines, the number of strokes of the pumps, and the amount of water pumped. The last item is indicated in the diagram by the length of the full black column—the third column in each space. Above the top of the black column is the outline of an addition to its height, intended to show the loss of water due to the slip or defective efficiency of the pump.

In order to give further indication of the process leading up to the decision, it is desirable to refer to the several points to which specific attention was paid, and with regard to these to record the following notes:—

1. The design of each tower was examined. Its practical stability was subjected to severe experimental test during the course of the trials, under winds which reached 45 miles per hour in one instance,—a very high velocity for an inland exposure.

Price has been taken into account in relation to the amount of water which can be delivered for a given outlay. The price is to a considerable extent determined by the size of the engine.

In the efficiency of delivery No. 3 stands conspicuously at the head in this respect. No. 17 comes next, and loses not a little by the inefficiency of the pump; then follow No. 8, No. 7, No. 14, and No. 16, in the order named. The inefficiency of the pump of No. 14 placed it at a considerable disadvantage in this respect.

With regard to the delivery of water, it should be remarked that, as may be inferred from its design, No. 3 delivers water equally for practically every part of the stroke. There is hardly any idle or slack time in the revolution of the wheel, and consequently the engine works with a very long pump stroke, and does a great deal of pumping for very few revolutions.

The pumps of the six engines were double acting pumps, or bucket and plunger pumps, with the exception of that belonging to No. 7, which was single acting. The conditions of the competition were favourable for the use of double acting pumps.

In forming a judgment upon the cost of maintenance in respect of repairs, regard has been had not only to the excellence of design and workmanship, but to the probable wear and tear under what may be called "agricultural" conditions. In this respect the number of revolutions of the wheel and the number of strokes of the pump for a given quantity of water delivered are very important items, as in ordinary wear it is upon these two that the life of the engine or pump depends. In both these particulars No. 3 is conspicuously successful. Then follow No. 16, No. 7, No. 14, No. 17, and No. 8. No. 7 is distinguished by the uniform thoroughness of execution of the design.

The ease of erection, which is an important item from the agricultural point of view, as it involves serious considerations of expense, has been estimated partly from an inspection of the designs and partly from experience on the ground. On the whole, No. 7 is regarded as entitled to special mention in this respect, and No. 17 presented obvious elements of disadvantage from this point of view.

Considering Table III. and Fig. 9, we get the performance of the mills. No. 3 is

clearly ahead of all the others in nearly every point, and easily takes the first prize. The points worthy of special commendation are—

1. Its general excellence of design, especially as regards the engine and pump.
2. Its efficiency as determined by the amount of water pumped.
3. Its successful governing.
4. The arrangement for the automatic application of the brake.
5. Its economy in upkeep, due to the slow motion of its moving parts, and its good workmanship.
6. Its reasonable price.

For the Second Prize a similar prominence in so many points is not to be expected, but when the qualifications under the several points enumerated are all taken into account, No. 7 stands distinctly ahead of the remaining four. The points for special notice with regard to this mill are—

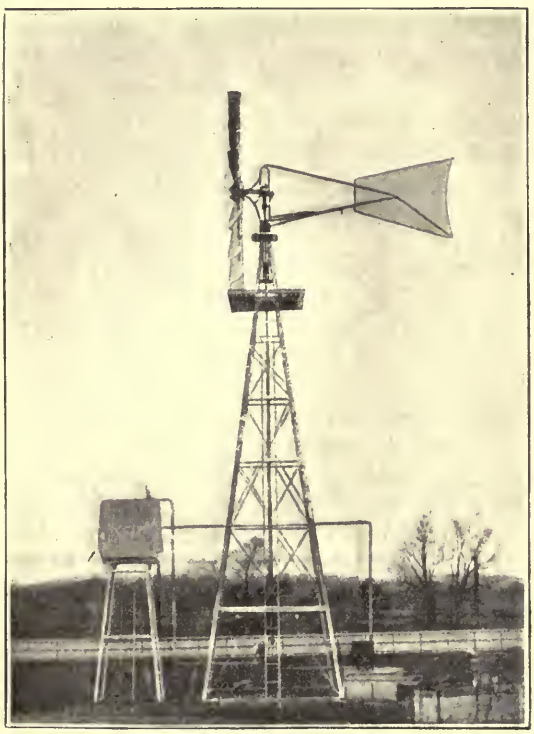


Fig. 10.—No. 3 Windmill.

1. The generally satisfactory nature of the design of the engine and of the tower, and the thoroughness of the workmanship in its execution.
2. The provision of an efficient brake.
3. The comparative economy of upkeep arising from the soundness of the design, and the comparatively slow motion of the pump.
4. The ease of erection.
5. Its reasonable price.

There was considerable diversity in the details of the sails, their shape, angles, and areas, and the trials were not conclusive as to the best form ; for some were evidently good mills, but handicapped by the conditions. However, the form of the sails, their angles, and dimensions of the prize winner are such as the experience in turbines generally would lead one to adopt. We will now briefly describe the final competitors, beginning with

No. 3.—*Messrs. Goold, Shapley, & Muir Co. Ltd.*, Brantford, Ontario, Canada.

This Canadian engine, illustrated in Figs. 10 to 12, has a wheel 16 feet in diameter, with 18 blades grouped in six sections, having an area of 131.32 square feet, with an available clearance area of 67.93 square feet between the blades and 30.88 square feet at centre of wheel.

The blades are fixed after the American fashion of threading them on the outer ring and fixing them thereto by means of a stamped steel bracket riveted to the sail and ring, the inner end of the blade being riveted to the inner ring. This method probably gives a maximum of strength for a minimum of weight, but it is open to the objection that should the middle blade of any one section of the wheel fail, in order to dismantle and replace it it is necessary first to cut off the adjoining blade. The wheel is mounted on a horizontal shaft running in two roller bearings of 6½ inches in length. The boss of the wheel is made very deep in order to allow of the effectual

bracing of the arms, but in order to avoid consequent increased overhanging the boss is recessed at the back and the roller bearing off the main shaft is carried well forward into it.

The gear of this engine differs from all others. The object to be attained was to get as long a stroke of pump as possible, a pinion on the main shaft gears into a mangle rack of such a length as to give one stroke of the pump shaft to two and a half of the wheel.

This gear is illustrated in Fig. 11, B, C, and D. It will be seen that there are two vertical parallel racks, with a connecting semicircular rack top and bottom, in which the pinion alternately engages; the rack is alternately thrown over at the end of each stroke by means of cams, thus reversing the direction of the travel of the pump rod.

The rack is guided between four steel rollers, which at either end of the stroke engage the cams; and to ensure the even working of the pinion in the rack a steel guide plate working against a flanged roller is provided.

The swing of this mangle action is only $1\frac{3}{4}$ inch, consequently there is very little angular motion of the pump connecting rod. The pump rod, which is of 3 inches square white maple, is connected to the rack casting by means of a piece of $1\frac{3}{4}$-inch wrought-iron pipe guided through the tower head.

The pump is a double acting syphon pump, 4 inches diameter by 22-inch stroke, the working barrel being of gun-metal and the valves of vulcanised indiarubber.

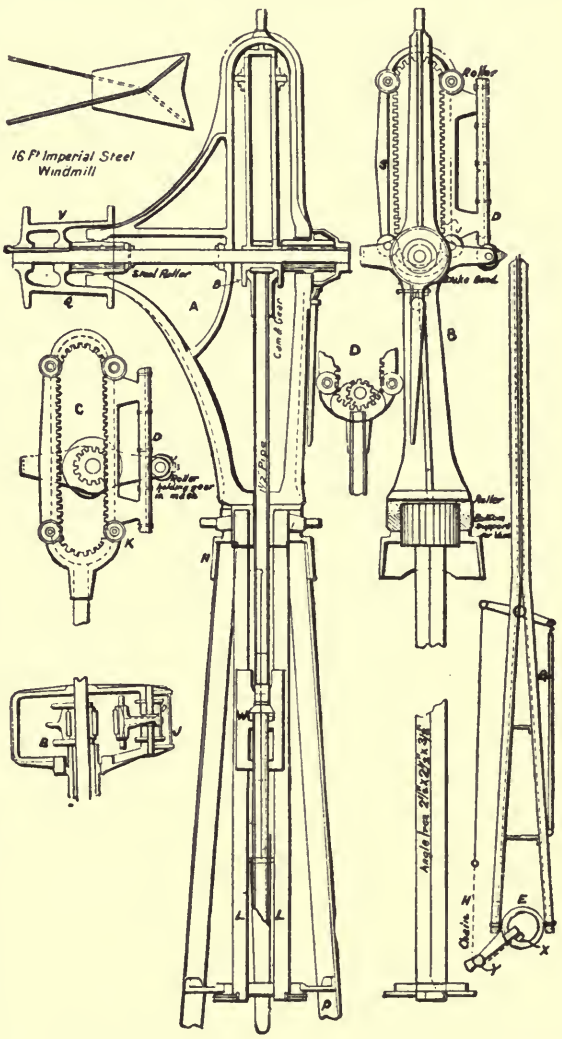


FIG. 11.—Details of No. 3. The Prize Winner.

The main feature of this engine is the method adopted of governing, which differs entirely from that of any of the other engines. This is shown in Fig. 11, E. On the frame of the wind vane a lever is pivoted, one end being controlled by a spiral spring anchored to the lower frame of the vane. To the other end of the lever is attached a chain H, connected with the pull-in wire, the tightening of which is done in opposition to the force of the spring and pulls the wheel into the wind. On freeing the pull-in wire, or in the event of the breaking of same, the spring pulls the wheel over parallel with the vane; consequently the edge of the wheel only is facing the wind; at the same time the brake is put on and the mill stops.

The tower consists of four angle steel posts, with five intermediate angle iron frames and diagonal tie rods. At the bottom of each post are anchor plates, which are

bolted to the timbers let into the ground about 5 feet deep. The general design and workmanship of this engine leaves little to be desired.

This engine was started on Monday, March 2, and ran most satisfactorily all through the trials. At the end of the first month it was one of the selected six, and ultimately gained the First Prize.

No. 7.—*Messrs. Thomas & Son*, 64 Broad Street, Worcester.

This engine is illustrated in Fig. 14. The wheel is 16 feet in diameter with 24 blades, having 141.55 square feet area, with an available clearance space between same of 55.5 square feet, and 23.04 square feet area at centre of wheel. The blades in this wheel are separately riveted to brackets fastened to the rings of the

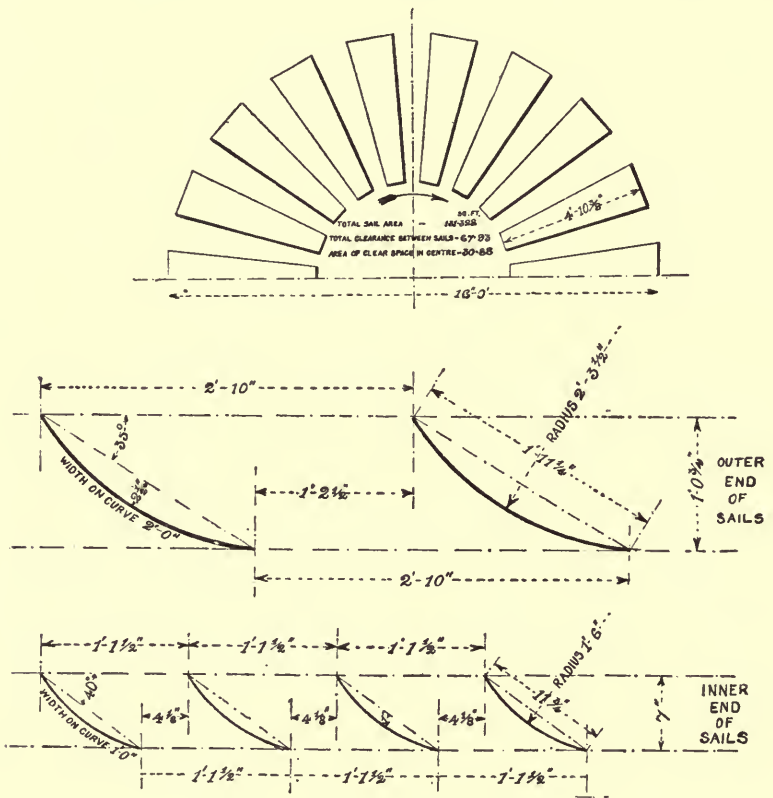


Fig. 12.—No 3. Details of Sails.

wheel, so that any one blade may be easily removed and another fixed in its place. The outer end of the blades are very materially strengthened by having two corrugations, which are seen in the illustration. The wheel is further stiffened by six stay rods from the outer ring of the wheel to a casting on the projection of the wheel spindle.

Immediately behind the wheel is an automatic band brake, which is actuated by a projection on the tail vane when it is blown over by a gust of wind, or when the wheel is pulled out of the wind by the windlass.

The wheel spindle is carried in two horizontal roller bearings of ample proportions, and the thrust of the wheel is taken by ball bearings at the end of the spindle. The tail vane itself is mounted on ball bearings. On the wheel spindle a pinion is keyed, which gears into a wheel on the pump crankshaft, speeded so as to give one

stroke of the pump for two and a half revolutions of the wheel. The teeth of these wheels are all tooled, being cut out of the solid. The cast-iron revolving head is carried on ball bearings at the cap of the tower, and a cylindrical sleeve descends through the tower head and is guided by four rollers. This gives a very efficient support to the head. The pump rod is of 1-inch wrought-iron tube, the weight of which is counterbalanced by a vibrating lever and weight.

The tower consists of four angle posts inclined approximately 1 to 5, with angle iron stiffening frames every 5 feet, and $\frac{5}{8}$ inch diagonal bracing rods. Placing the stiffening frames at the intervals named materially facilitates the work of erection, as the tower forms its own scaffold as the work proceeds.

The pump is of the single acting type, fitted with gun-metal bucket and suction valves.

The design and workmanship both in the engine and tower were most carefully thought out and executed, and throughout the trial the engine worked very satisfactorily, and was ultimately awarded the Second Prize.

No. 8.—*Mr. John Wallis Titt*, Warminster.

The wheel of this engine is 16 feet in diameter, sail area 113.89 square feet, clearance area between blades 74.59 square feet, and 38 square feet area at the centre of the wheel.

The stroke of the pump can be varied. The wheel is mounted on a crankshaft with roller bearings. The head is built of steel angles and plates, and revolves on roller bearings.

Another point in this mill is the provision of a ladder revolving with the head gear, so that a man can without danger lubricate or attend to other small matters in connection with the gear while the engine is running.

The wheel is controlled by a tail vane and weighted lever. This lever is raised by the starting wire, which passes up through the centre of the mill and over two small guide pulleys. During the trial the wire failed, and the absence of any brake on this gear was very apparent. The general arrangement of the head is shown in Fig. 13.

Fig. 13.—No. 8. Mr. John W. Titt's Mill.

The pump is of the bucket and plunger type, $3\frac{1}{4}$ inches diameter and $6\frac{1}{8}$ inches stroke, and there being no intermediate gear the efficiency of the pump was good. At times, however, the speed of the pump is high, and unless the plunger is properly lubricated it would tend to wear more than a slower running pump.

No. 17.—*Messrs. Henry Sykes Ltd.*, 66 Bankside, London, S.E.

This engine presents several novel features which had evidently been carefully thought out.

Unlike other engines having fixed blades in their wheels, in which the tendency is rather to increase the size of the blades and to diminish their number, the opposite course is adopted of diminishing the width and increasing the number of the sails, on the ground that most work is done by the wind on the leading edge of the blade; and consequently it is well to have as many leading edges as practicable. The periphery of

the wheel has a hoop round it, to which the end of the blades is fixed, which gives it a very strong and, at the same time, neat appearance. Another novel feature of the engine is the jointed tail vane, described later.

The wheel is 16 feet diameter, with 42 blades 4 feet 6 inches long, 9 inches

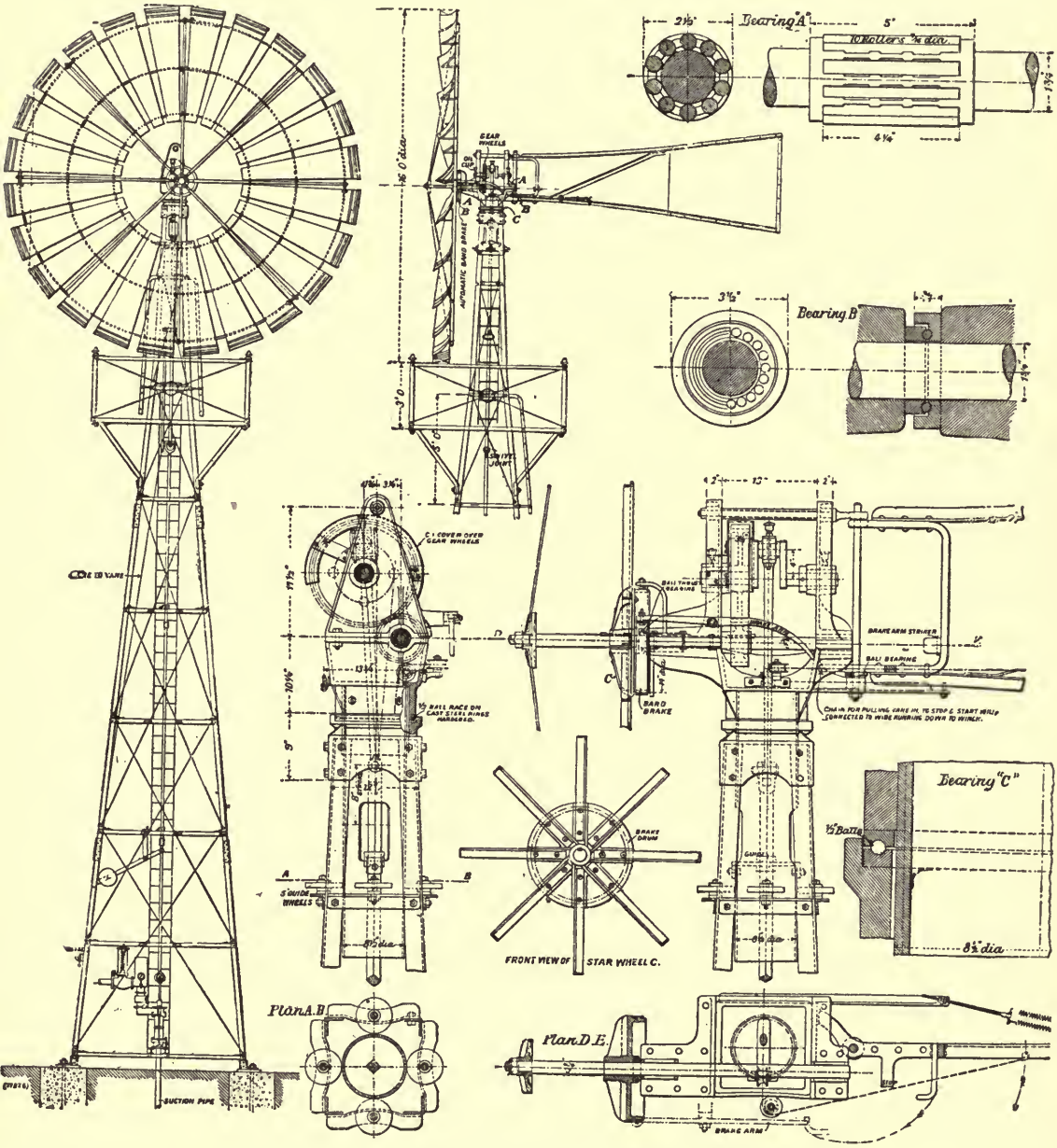


FIG. 14.—No. 7. Messrs. Thomas & Son's Mill. Second Prize.

wide at the periphery, and 6 inches at the inner end. The sail area is 142.82 square feet, with 54 square feet available clearance between them, and 38.48 square feet at the centre. The curvature of these blades is shown in Fig. 16. The wheel is mounted on a plain single-throw crankshaft, carried in an inclined position in two gun-metal bearings in the revolving head, the thrust of the wheel being taken by a

collar on the shaft, bearing against the side of the main gun-metal bearing. The head revolves on roller bearings.

The tail vane is mounted on two rods attached to the revolving head, which always tends to make the wheel face the wind, the inclined hinged axis of vane, a fixed portion of tail vane, and a side vane, which tends to make the wheel face away from the wind and only present its edge to it.

Diagram Fig. 15 shows the position of the wheel and vanes when the engine is running in a moderately light wind, the wheel facing the wind.

We need not describe more of them from the report, but may notice the head gearing of the "Samson" Mill by Messrs. J. S. Millar & Sons, Annan.

This "Samson" mill is very similar to the "Ideal" mill; the wheels are the same, but the gearing is somewhat different, in that two small pinions on the wheel shaft gear into two wheels with a pin between, forming the crank for the pump rod, taking the place of the annular gear in the "Ideal" mill. The governing arrangements are the same in each. The "Samson" tower is of the same design, but is a 4-post tower instead of tripod. The gearing is illustrated in Fig. 17.

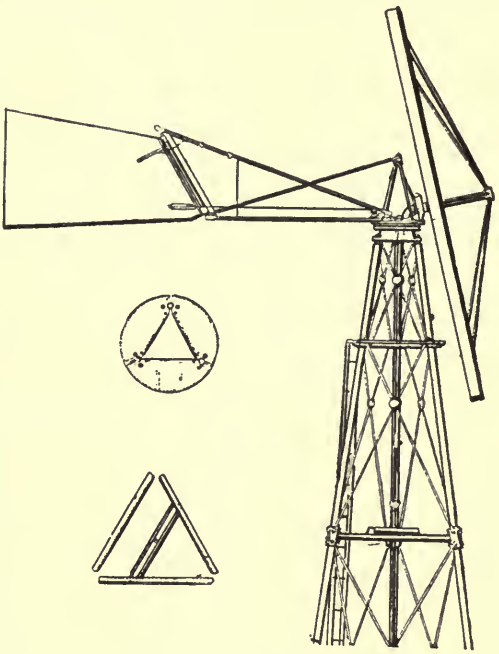


FIG. 15.—Details of No. 17. Henry Sykes' Mill.

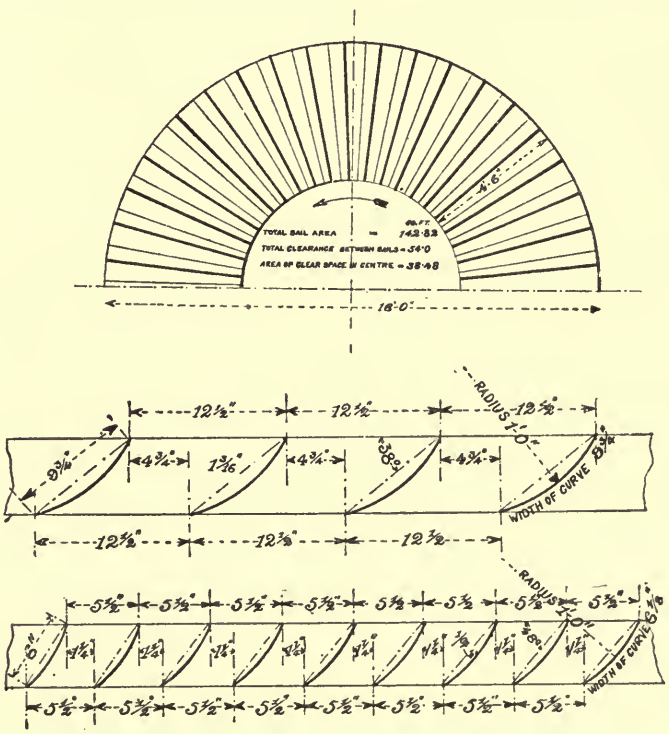


FIG. 16.—No. 17. Henry Sykes' Mill. Details of Sail.

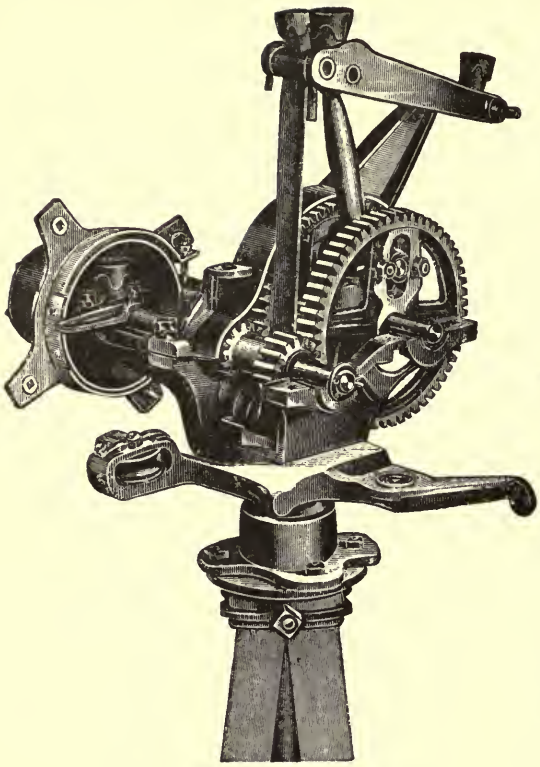


FIG. 17.—Windmill Head Gear.

An exceedingly simple little windmill (Fig. 18) may be constructed of two half discs of sheet metal, zinc for preference, or aluminium. The sails are at right angles to each other, and the shaft passes between at an equal angle to each sail. It works best with the shaft parallel with the wind. The half discs are tied to the shaft and to each other by steel wires, as shown. It has been called by a fanciful name, the Pantanemone.

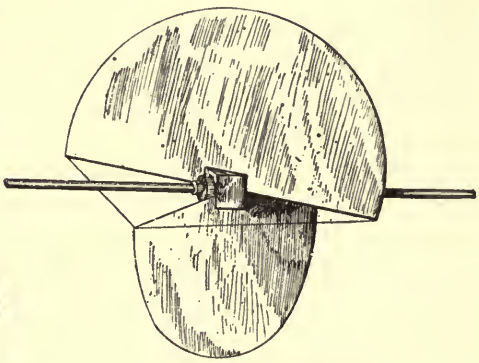


FIG. 18.—Simple Windmill.

HYDRAULIC MACHINES

The turbine wheel has completely superseded the water wheel. For all falls of water the turbine can be designed to give better results.

In Switzerland and in the United States of America water powers to great extent are used. In 1880 there were 55,000 water wheels working in North America, with a total aggregate power of 2,000,000 horse-power. Since then this has been largely increased by the introduction of electrical transmission, so that the horse-power is now considerably over 2,000,000.

At Geneva water power to the extent of over 7000 horse-power is derived from the Rhone, and 12,000 horse-power more from the same river about 5 miles farther down the stream.

In the British Isles there are no great falls of water, but in the more hilly and mountainous districts water powers could be developed of quite considerable amounts in Wales, Ireland, and North Scotland. Coal, however, is cheap and plentiful, so that the available water powers are neglected.

Water power has been estimated to cost in America about £4, 10s. per horse-power per annum, and steam power from coal at 8s. a ton costs about the same. This, however, is for low falls ; for high falls the cost is much less, and in some falls over 100 feet it has been as low as £2 per horse-power per annum.

The first cost of water power varies very much with the district and conditions, and cannot be laid down on any fixed basis. The large works at Geneva cost £30 per horse-power, the smaller works £60 per horse-power.

With water power the bulk of the cost is due to rental of land and water rights,

interest on capital and depreciation of plant, only a small fraction for working expenses; with steam power about half the cost is due to wages and fuel, and the other half to permanent charges.

A steam plant therefore costs less when idle, whereas in a water power plant costs go on all the time full. If the water cannot be used night and day the whole cost falls upon the hours during which it is used.

Storage of water is of great importance in all water powers, and in some cases it is the simplest and best means for storing power to pump water up to a high reservoir; and in some cases hydraulic accumulator storage can be utilised where a high level reservoir is not available. Lord Armstrong's high pressure hydraulic distribution of power system is a good example of this method.

The hydraulic accumulator is simply a tall cylinder with a heavily loaded ram, as shown in Fig. 19. The water is pumped in continuously in small quantity, and maintained under the pressure of the load on the ram, the ram rising and falling as

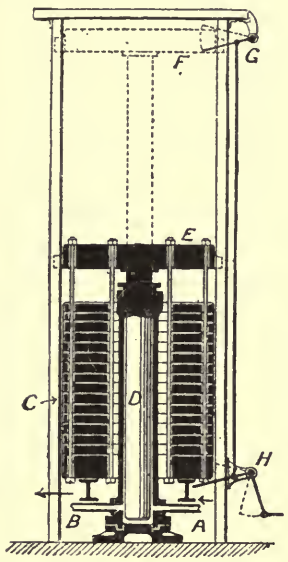


FIG. 19.—Hydraulic Accumulator.

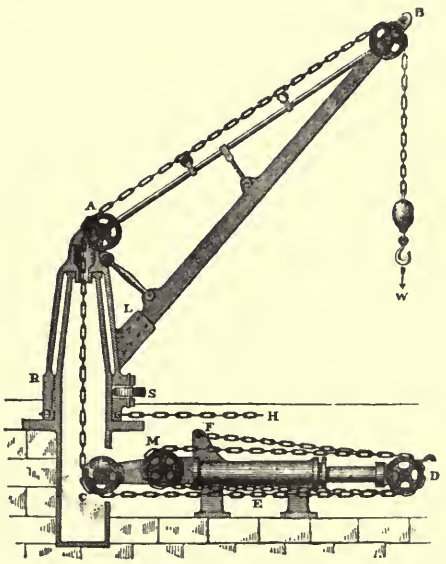


FIG. 20.—Hydraulic Crane.

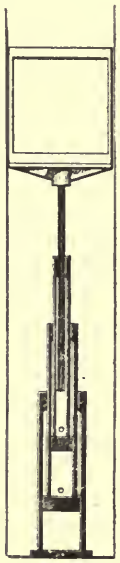


FIG. 21.—Hydraulic Telescopic Lift.

the pressure water is used up or stopped. If a is the area of the plunger or ram, and P the load upon it, then $\frac{P}{a} = p$, the pressure per square inch on the water delivered in lbs. If L is the length of the cylinder full of water, then aL is the maximum storage, and the energy stored in foot-lbs. is equal to the quantity of water stored multiplied by the pressure per square foot.

Thus one of the accumulators in the London Hydraulic Power Company's station, working at a pressure of 100,000 lbs. per square foot with a 20-inch ram, $a = 2.2$ feet, and a stroke of 23 feet, the energy stored is only $\frac{2.2 \times 100{,}000 \times 23}{33{,}000 \times 60} = 2.5$ horse-power for an hour.

But it could deliver this energy very quickly. In one minute it would work at the rate of 150 horse-power.

It is a costly storage apparatus, but for the purposes to which it is applied its great rate of discharge for a short period makes it worth the money. This form of hydraulic storage is used largely for lifts and cranes, and an excellent but small system exists in

Glasgow and London very successfully. At Birkenhead Docks and Woolwich Arsenal hydraulic cranes are so worked. A simple hydraulic accumulator will work a number of cranes and hoists.

In hydraulic cranes the ram carries a sheave of pulleys with chains for multiplying the motion, as shown in Fig. 20. The chain is fixed at F, passes over one pulley at D on the ram, then round fixed pulley M, and again over D, and from thence over guide pulleys C, A, B to weight W. A chain H driven by a separate ram is used for slewing the jib.

The hydraulic lift as now most commonly made has a telescopic ram, one being fitted inside of the other as shown in Fig. 21, so that when closed down only a shallow well beneath is necessary to accommodate a high lift. Thus with one cylinder and three concentric rams, each lifting 10 feet, a height of 40 feet may be obtained. In Richmond's patent differential telescopic high lift hydraulic lift the water under each piston is forced into the next cylinder above, so that the rams all travel up simultaneously at same speed, all reaching the top at same time. This is by far the best and safest lift, and is the only one to be recommended for heavy passenger work.

WATER PRESSURE ENGINES

These are of two classes, reciprocating pumping and blowing engines and engines with rotatory shafts and cranks. In the design of these engines it must be borne in mind that water is a heavy inexpansible fluid, which has considerable inertia, reacting on anything which it meets to divert its course of motion. Many water pressure engines have been designed without regard to this elementary fact, and with certainty of failure.

The simplest form is that designed to work a bellows or pump by a to-and-fro movement only. The more complex forms are those producing rotary motion. A moving column of water has velocity and pressure, and head or elevation, *i.e.* the difference in level between the place where it starts from to the place it falls to.

The total energy is the sum of its momentum, its pressure head, and elevation head. If V is the velocity in feet per second, p = pressure in lbs. per square foot, h = the elevation, and g = the weight per cubic foot, the total energy per lb. of water is equal to $\frac{V^2}{2g}+\frac{p}{g}+h$ in feet.

All hydraulic engines are arranged to work with one or other of these quantities. For example, in an old overshot or breast wheel the water drops into the buckets at the high level with as little force as possible, only falling a few inches into the buckets; it then merely acts by its dead weight on the loaded side of the wheel; the energy is simply proportional to $h \times w$, that is, height of fall multiplied by the weight falling. A cubic foot weighs 62.5 lbs. roughly, and, falling in a bucket a height of 10 feet, would give $62.5 \times 10 = 625$ foot-lbs.

The pressure head is used in all piston engines. Here the water flows from the high level at a slow speed through a pipe, the pressure on the piston is that due to the height of the column of water; thus a 10-foot head would give a pressure of 625 lbs. per square foot on a piston at the bottom; thus we can utilise p, the pressure.

In the third method we utilise V, the velocity. In this case the water is allowed to flow with the full velocity due to the pressure, and to strike against a free moving blade or vane, which arrests its motion and turns the jet aside. The energy at any velocity V is $\frac{mV^2}{2g}$. If we take a 10-foot fall, again, we find the V for this fall to be equal to $8\sqrt{h}$ $= 8\sqrt{10} = 25.2 =$ velocity, and $\frac{V^2}{2g}$ = energy per lb. Hence $\frac{25.2^2}{64.4} = 10$ approximately, the energy per lb. of water, so that the cubic foot would have $10 \times 62.5 = 625$ foot-lbs. of energy. It will be seen, then, that we get the same amount of energy out of every cubic

foot of water at the same fall whether it acts by h, its dead weight on a wheel, or by pressure p on a piston, or by velocity V on a vane or movable blade.

In some water power engines we have p and V acting together, as in Professor James Thomson's vortex turbine.

In the water pressure engines now under consideration p is the acting force, V being kept below 2 or 3 feet per second by using large supply pipes and ports.

Hydraulic pressure engines are only to be used for very high pressures and for slow,

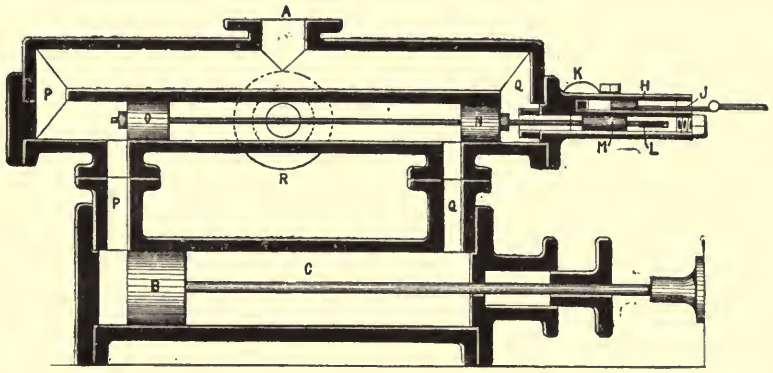


FIG. 22.—Hydraulic Engine.

steady motion. The volume of water used per stroke by pressure engines is constant, however much energy is exerted. Water can be saved only by reducing the number or length of strokes on small load. The efficiency at full load is about 80 per cent., at half-load under 35 per cent. To get over this difficulty many devices have been proposed, such as the use of two or more cylinders in cranes or lifts, so that for light loads one cylinder is used, and for full load all the cylinders are used.

For pumping water the load is constant, hence a simple single cylinder is used, and also for bellows blowing, such as shown in Fig. 22.

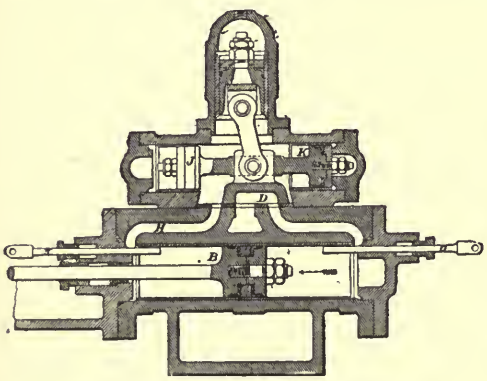


FIG. 23.—Hydraulic Engine.

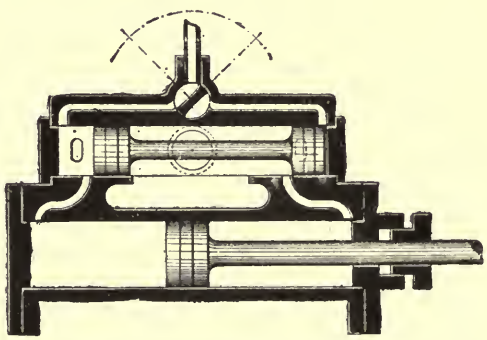


FIG. 24.—Hydraulic Engine.

The first successful engine of this type was probably that of the design by Mr. Joy, who applied the auxiliary valve. There being no rotary motion, the valves must be operated in the first instance by some tappet motion, like that shown in Fig. 23, in which the piston B makes the little valve rods projecting through the cylinder cover move when it comes to either end of its stroke. Now, if these rods are made to actuate a slide valve like D direct, there being no momentum and no expansion, as soon as the valve opens very slightly the piston stops and the valve ceases to open; hence only a small amount of water flows and the speed is restricted.

The idea of the auxiliary valve is to provide a small easily worked piston to which the tappets admit water under pressure. This auxiliary piston, having only to move the D valve, moves easily and quickly, and being balanced by a piston or other means moves with little friction, and opens the ports full.

Another form with a rotary auxiliary valve is shown in Fig. 24, the valve being operated by a tumbler motion which falls over and opens the valve wide.

In organ blowing engines the auxiliary valve may with advantage be operated by air pressure from the bellows and started by a hand valve.

The rotary shaft engines are usually oscillatory, thus obtaining a very simple valve mechanism. One which has been much used is that shown in Fig. 25, the Schmidt engine. The oscillation of the cylinder opens and shuts the ports ; the cylinder is carried on trunions E, concentric with the curved faces of the valves. These trunions are carried on stiff levers pivoted to the bed, and by means of screws can be depressed so as to keep the valve faces close up as they wear away. An air vessel is also provided on the supply to give some elasticity and prevent shocks.

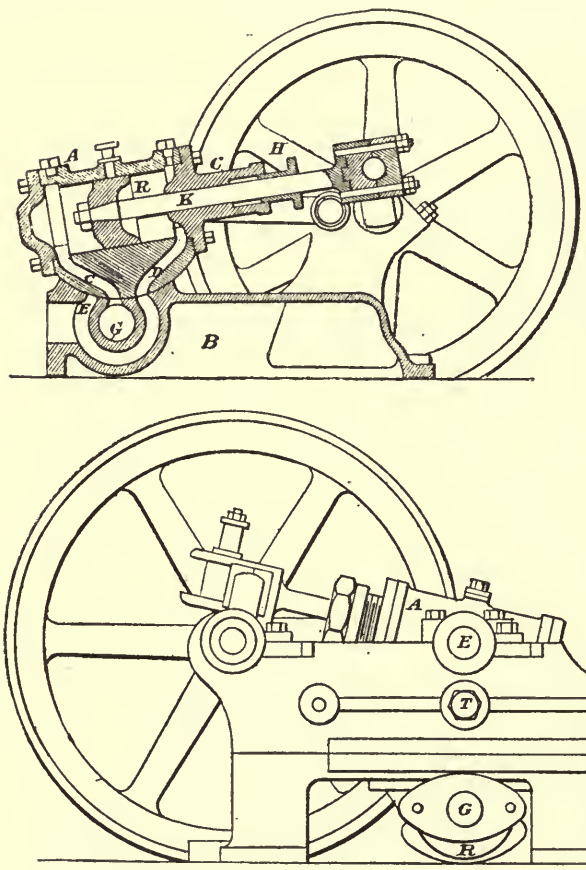


Fig. 25.—Schmidt's Hydraulic Oscillating Engine.

In Hastie's engine we have an example of an effort to construct a 3-cylinder engine with a variable piston stroke, so that the water used will be more in proportion to the work done, and it met with some degree of success. The crank pin is moved on a slide against the tension of springs, so that the greater the resistance offered by the load the longer becomes the centre of the pin from the crank shaft centre, and hence the greater the leverage of the piston and the longer the stroke.

In the same direction of improvement a later type of engine has been developed by Mr. Arthur Rigg with four cylinders. In Hastie's engine the cylinders were oscillatory, but otherwise stationary while the crank revolved. In Rigg's engine the crank pin does not revolve, but is capable of a lateral movement horizontally, while the cylinders revolve round the pin and also oscillate. The plungers are pivoted to a revolving crank disc, as shown in Fig. 26. If the central hollow pin, which also acts as valve, is central, then the engine stands still, and if moved to one side excentric ; the length of the stroke is twice the eccentricity. The central pin is carried on differential hydraulic plungers, by means of which it can be moved horizontally to right or left. There are two ports, an exhaust and pressure port, on the central boss, which is faced to press against the fixed ports, and each cylinder receives water during one-half of a revolution and exhausts during the other half.

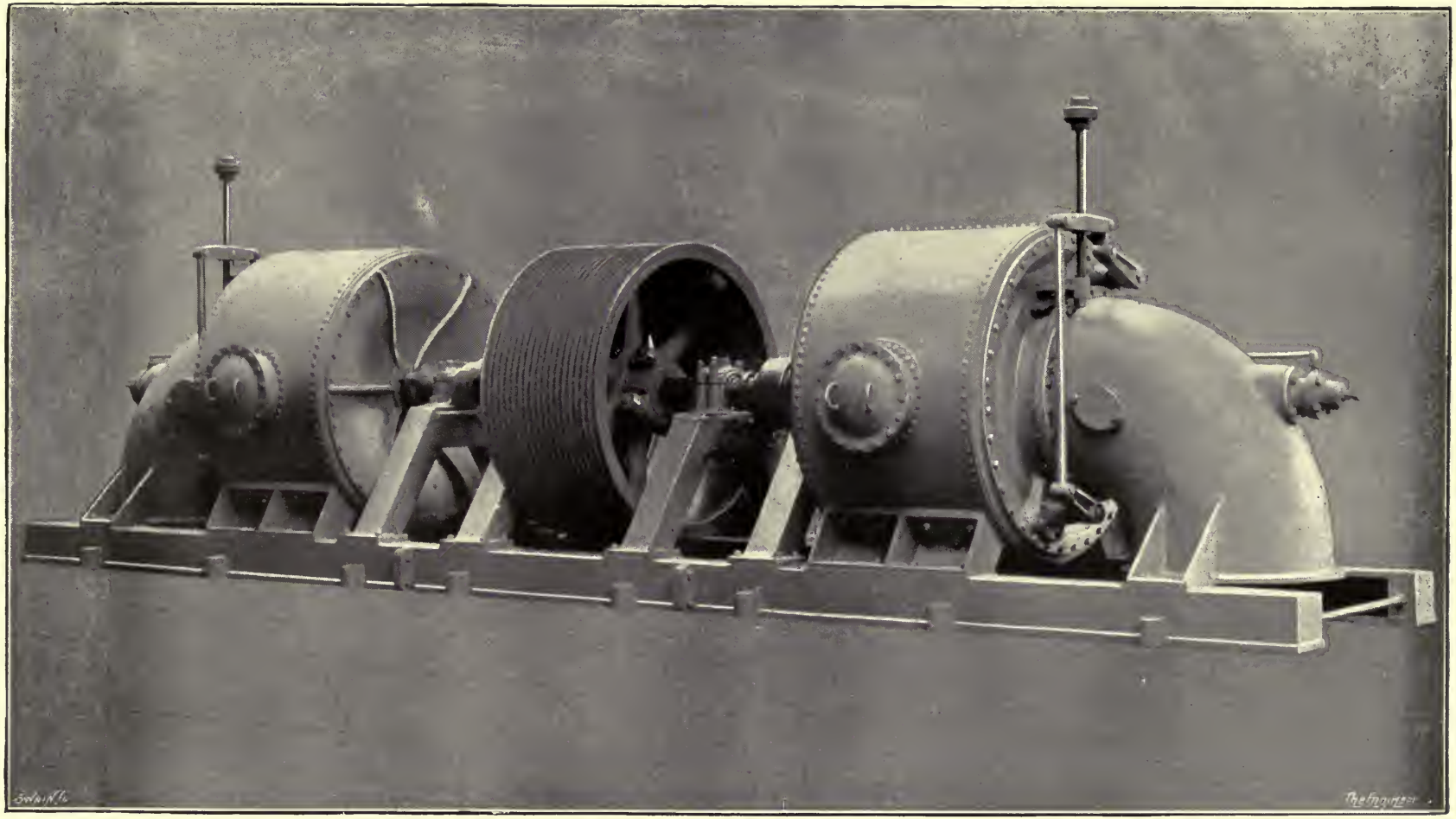


PLATE I.—GUNTHER'S TWIN VORTEX TURBINES.

If the pin is moved to one side the disc revolves in one direction, and if moved to the other side it revolves in the opposite direction, and the stroke and consumption of water is adjusted to the work to be done.

In Hastie's engine the adjustment of stroke is automatically proportionate to the load; the Rigg engine requires to be adjusted by hand by valves to the load.

The water enters by pipe on the left hand, and by the hollow plunger reaches the plunger ports. The water pressure in the open-ended hollow ram tends to move the ram to the right, while the right-hand larger plunger can be moved to the left by admitting water pressure; the water is admitted to or released from the larger plunger by two valves, so that it can be adjusted at any point, and by shutting both valves the plunger is locked in position.

A small engine with $2\frac{1}{4}$-inch rams working at 700 revolutions per minute at 500 lbs. pressure runs well, and has power enough to work a heavy crane.

These and other hydraulic pressure engines have a very limited sphere of usefulness, although of much importance wherever they can be used, such as for intermittent work. The high pressure of accumulators is required to give them power, and the internal strains are very great, so that they are heavy machines and somewhat costly.

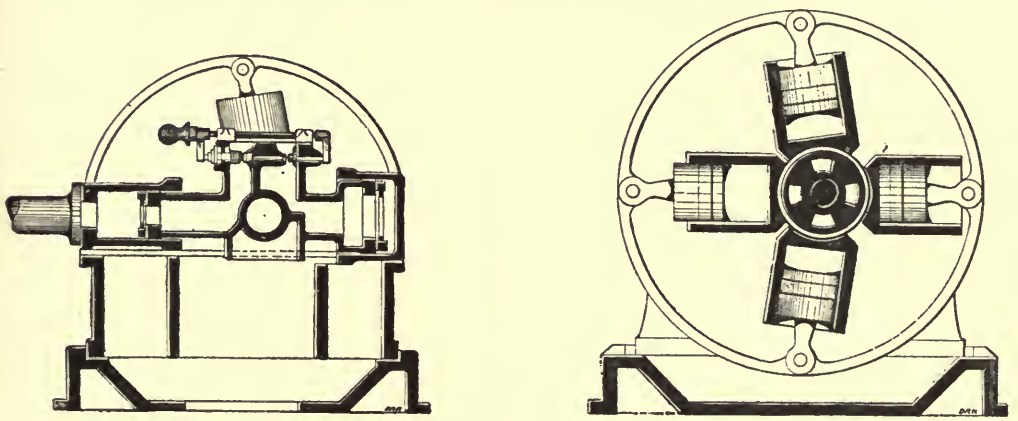


FIG. 26.—Rigg s Engine.

An hydraulic system for working hydraulic cargo lifts and cranes is exceedingly useful on steamships, as they work silently, efficiently, and with a maximum of safety, and do not create the intolerable clatter and noise raised by the steam winch.

WATER TURBINES

This class of hydraulic machinery is the important one. They have been roughly sorted into classes according to the direction of the water flow in relation to the turbine wheel. As a matter of fact, it makes little difference whether the flow is in one direction or another; but the classification serves as a brief description of the general arrangement of the machine.

We have, 1st, axial flow turbines, in which the water flows parallel to the axis of the wheel.

2nd. Inward radial flow turbines, in which the water flows in radially from the outer to the inner periphery.

3rd. Outward radial flow turbines, in which the water flows from the inner to the outer periphery.

4th. Mixed flow turbines, in which the water flows radially and axially.

5th. Free jet turbines or tangential flow turbines, in which a jet or jets of water strike the buckets or vanes tangentially.

But the real fundamental classification of turbines divides them into two classes only—

1st. Pressure turbines.
2nd. Impulse turbines.

The first act partly by the pressure of the water and partly by its momentum, that is, by V and p. When the water enters the turbine it has pressure, as all the energy of the fall has not been developed into velocity. It has velocity also, which it loses in passing through the turbine, giving up the corresponding energy in doing so.

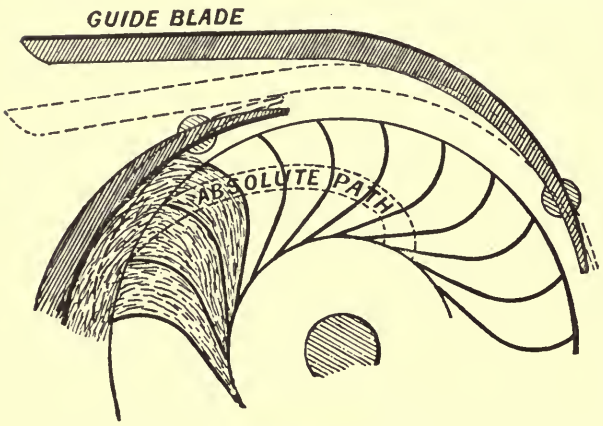


FIG. 27.—Diagram of Pressure Turbine.

As there is pressure on the water as it enters it must fill the whole turbine case, which must be water-tight except at the outflow; the water must act on the whole wheel.

In the second class the whole head of water is developed into a jet or jets of great velocity, due to the fall; such jets have no side pressure. This may be seen by simple inspection, for a free jet moves for a distance from the orifice as a straight rod of water, not expanding sideways as it would do under pressure.

Water under pressure transmits that pressure sideways, forward, and in every direction; but in a column of water moving at the full velocity due to the head there is no side pressure, the whole energy is in the forward movement.

Pressure turbines work best when drowned, so that no part of the fall is lost. But they will work equally well up to 25 feet above the tail race if the outflow pipe mouth is carried down below the tail water surface, this part of the fall being utilised by the suction of the water falling down the pipe. Acting like a syphon, this outflow tube is called a draught tube. This facility is of value in many cases, for it is not always advisable nor possible to have the turbine at the tail water level. And when the tail water level is variable in height this draught tube allows of the rise and fall of the level without affecting the turbine, except to reduce the suction when the tail water level rises. Whereas the impulse turbine must be placed above the highest level of the tail water, it will not work drowned; hence the whole fall cannot be utilised at low water level in the tail race.

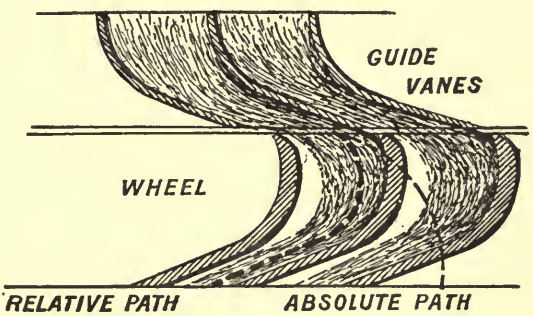


FIG. 28.—Diagram of Impulse Turbine.

Some turbines have been designed to work as impulse and pressure turbines, so that when the tail water is low they work by impulse, but when high and the wheel drowned, work as pressure turbines. Such turbines are called "limit turbines." They are not successful; it is by far better to use a pressure turbine with a draught tube long enough to keep the wheel above high water mark.

The two forms of turbines may be illustrated in diagrams (Figs. 27 and 28). Fig. 27 is a pressure turbine inward flow, with guide blades so as to give direction to the water and regulate the quantity; one guide blade and three wheel passages are shown filled with water. The water under pressure enters the wheel tangentially, and its course is

reversed in the passages through which it flows and falls in pressure, so that when it emerges it has no pressure.

The Jonval type of turbine, upon which system so many pressure turbines are made, is shown in Fig. 29 as a diagram. It consists of a ring of guide blades curved to guide the water into a ring of wheel blades curved the opposite way. The water presses

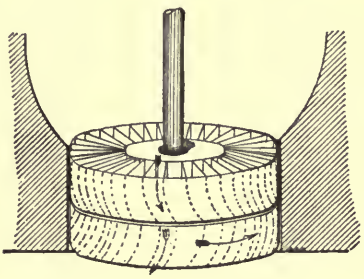


FIG. 29.—Jonval's Turbine.

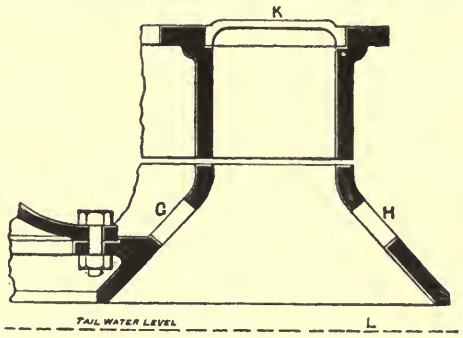


FIG. 30.—Impulse Turbine.

through the full wheel passages, strikes and presses against the wheel blades, losing its pressure in driving the wheel.

The impulse turbine is shown in diagram (Fig. 28) with two jets full and two wheel buckets nearly full. Here the water is spurted at full velocity against the vanes in the wheel, which are wide enough apart to allow the water to be deviated without filling

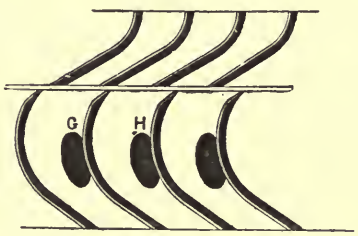


FIG. 31.—Impulse Turbine. Guide Vanes and Wheel Blades.

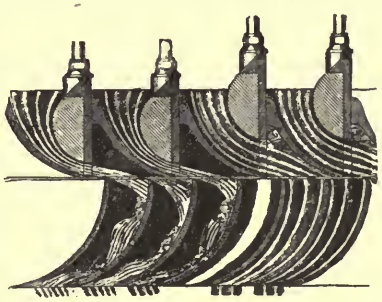


FIG. 32.—Fontaine Turbine.

the buckets; and in order to allow the water to spread out as it flows through the wheel, the vanes are splayed out towards the outfall as shown in the cross section in Fig. 30. The guide blades and buckets are shown in Fig. 31. K is the guide blade, G and H are vent holes in the side of the wheel buckets to allow air to escape and so prevent back pressure. In the impulse turbine there may be as many guide blades as buckets, and the water may flow full bore all round, but that is not necessary; there may be only a few water jets on a large wheel, in which case it is called a partial flow turbine.

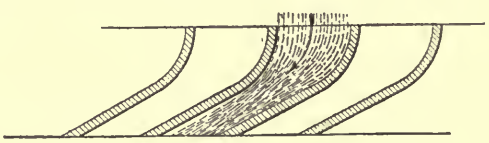


FIG. 33.—Blades of Fontaine Turbine.

In the old Fontaine turbine, shown in part section, the jets could be closed separately by sliding gates, as shown in Fig. 32. This shows in the two partially closed jets that the water does not flow steadily, but is broken up in the wheel, and so its power is wasted. Again, in the full flow the water chokes the wheel buckets (Fig. 33), because they are not splayed out at the exit, and the water requires pressure to force it out.

In Fig. 34 the Girard improvement is seen, wherein the buckets are curved and splayed so that the water at full gate never fills them, but leaves an open ventilating space.

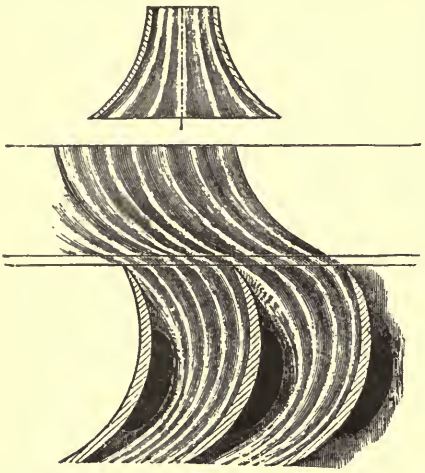


FIG. 34.—Girard Turbine.

In these impulse turbines, partially closing the jets is bad regulation; for regulating purposes the active jets should be reduced or increased in number by stopping them off one at a time.

The simplest impulse turbine is the Pelton wheel, driven by a free jet. This is shown in diagram in Fig. 35, the wheel carrying on its circumference a series of double cupped buckets; the jet at full velocity due to $\sqrt{2gh}$ strikes the mid-rib, splits into two, and glances off reversed in motion. Now, in reversing it exerts a great pressure on the cups delivering up its momentum.

In choosing a turbine we must be guided first by the condition of the tail race. If the tail water level is liable to considerable variations, then a pressure turbine should be selected to work either drowned or through a draught tube. But the impulse turbine is much easier regulated where water supply is variable and none too much of it.

Again, pressure turbines run up high in rotational speed on high pressures, while impulse turbines can be made with partial flow and with wheels large in diameter, so that with the same vane speed the rotations are smaller.

The pressure turbine is cheaper than the impulse on moderate falls with lots of water, but for the higher falls the impulse turbine, like the Pelton wheel, is cheaper and better.

To sum up the features of the two classes. The impulse turbine must always discharge above the tail water. It will work with complete or partial flow for full load. Regulated by closing one or more guide passages.

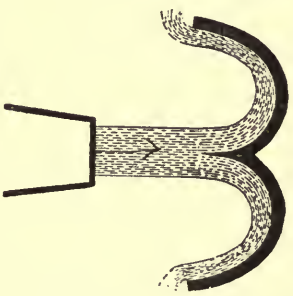


FIG. 35.—Pelton Wheel Bucket.

The pressure turbine discharges either below tail water or above tail water through a draught tube. It is regulated by closing some of the guide passages or by varying their area or throttling the water supply. At full load turbines are all highly efficient, some as high as 85 per cent.; but at lower speeds the efficiency falls off rapidly. The most efficient governor at low loads is that applied to the Vortex pressure turbine (Fig. 27), where the inlet to the wheel vanes is closed or opened by the guide blades moved by a governor.

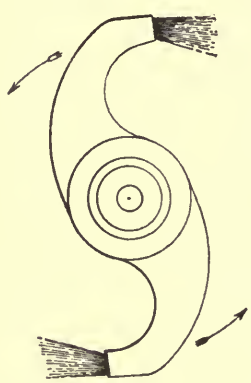


FIG. 36.—Reaction Wheel.

Here the impulse turbine has the advantage, as we can stop off the jets without causing losses. A Girard turbine of 200 horse-power on a test gave an efficiency of 0.8 full load and 0.802 on half load, the speed the same in both cases.

There is another type of turbine, not much used but of some interest scientifically, as it illustrates the true pressure turbine.

Two nozzles mounted on a hollow shaft, and pointing tangentially as shown in Fig. 36, will rotate with power when pressure water is supplied through the shaft. To understand this turbine we begin with the knowledge that a fluid water under pressure exerts an equal pressure on all sides of a containing vessel. Now, if we remove a part of one side of a containing vessel under water pressure the water will flow out of the

opening, but the total pressure on the side of the vessel in which the opening is made will be so much less than the pressure on the opposite side by the area of the hole multiplied by the pressure. Thus if the vessel had four sides of 9 inches area, and out of one side we cut a 1 square inch hole and the pressure per square inch is 10 lbs., then on the side out of which the inch hole is cut the total pressure will be $8 \times 10 = 80$ lbs. on that side, while the pressure on the opposite side will be $9 \times 10 = 90$ lbs. A difference of 10 lbs. pushing in that direction would move the vessel away opposite to the issuing water. And to obtain efficiency the vessel should move at a speed at least about $.8\sqrt{2gh}$.

The chief point to observe in the design of this class of turbine is that the supply of water to the vessel with the nozzle or hole is abundant and sufficient to maintain the full pressure behind the hole. And if the water is to pass through a hollow shaft this means a very large shaft.

But this can be overcome and a good construction made of a turbine on this principle. At the speed required for high falls the water in the wheel acquires an enormous centrifugal force, and the friction of the water is considerable. By making a compound wheel this speed can be reduced to half. It is of importance in steam turbines, this modification for reducing speed without loss of efficiency.

TURBINES, THEIR CONSTRUCTION

We shall now take up each class of turbine with a view to examine their construction. In the preceding division we have only considered their general mode of action and application. When one comes to consider turbines practically, not only is the direction of flow axial, or radial or mixed flow merely a matter of convenience, but there is little or no difference in action between a pressure turbine and an impulse turbine. We have followed this classification so far because it is the usual text-book classification. In the following descriptions we shall see that all turbines work by reaction, by entering a wheel in one direction, deviating in the wheel, and emerging as nearly as possible in the opposite direction. It is a familiar fact that any body, solid or fluid, moving in any direction and meeting another body which intercepts the moving one and reverses its movements, experiences a pressure proportional to the square of the speed before impact.

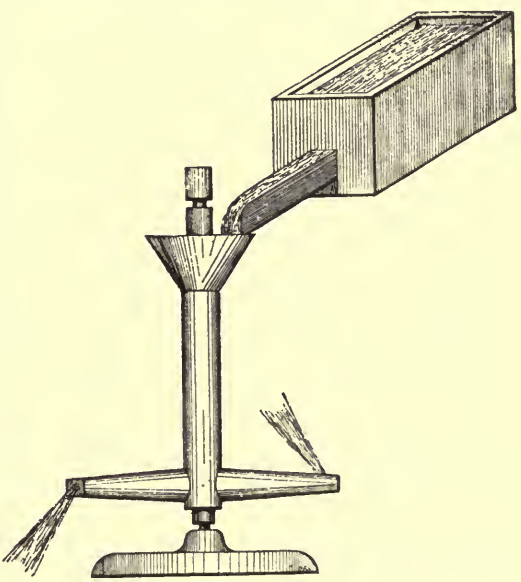


FIG. 37.—Barker's Mill.

In this respect all turbines are alike: the water must be deviated by the wheel. And so we best can select a starting-point from a turbine which can be dissected into either pressure or impulse wheels, observe the deviation of the water and the production of the acting pressures.

All turbines can be developed from Barker's mill, the original form of which is shown in Fig. 37, and which was merely a development of Hero's steam turbine invented 2000 years before.

Whitelaw and Stirrat, in Scotland, made a special study of this wheel, and concluded that the arms should be curved into an Archimedean spiral, which of course

would finish at a tangent to the periphery, as shown in Fig. 38. The black spiral line is the curve of the line of the arm. Whitelaw found that the arm should taper from the centre, beginning wide and gradually narrowing to the periphery as shown in the left hand curve, but later tests have proved that the opposite curve, widening out to the periphery, is equally as good and more efficient.

The two forms are shown better in Figs. 39 and 40, which show one with the narrow end of the nozzle outward and one with the wide end outward. Now, an inspection will show the difference in Fig. 40. With the narrow end n out the full pressure will be continued up to the very end, and the jet will issue at full speed, so that the wheel periphery must move back with a speed about $0.8\sqrt{2gh}$; but the point of pressure pushing the wheel back is down at b near the line F, and the pressure acts on the back of the nozzle at P.

FIG. 38.—Diagram of Curvature of Arms of Reaction Wheel.

Now, in the other case (Fig. 40) the full pressure is at line F, and the water deviates along the reversed curve all the way, and the pressure is distributed along the curve from a to b gradually without the water touching the other side of the nozzle P. Here the issue of the water from the narrow part is restricted by the curve a, b, and its outflow is at a less velocity than if it issued direct into the atmosphere as in the first case; consequently the efficiency is higher. Another way to look at it is to consider, in the first case the centre of back pressure is near the line F, while the centre of back pressure in the second case is midway between a and b, farther from the centre.

Now, if we cut the nozzles across at the line F and fix the centre piece with the one inner part of the nozzle, and mount the outer part of the nozzles on a rotatable shaft, Fig. 39 would exactly represent pressure turbine guide blades and vanes, in which the pressure would be due to reaction in b. And Fig. 40 would exactly represent the guide blades and wheel vanes of an impulse turbine.

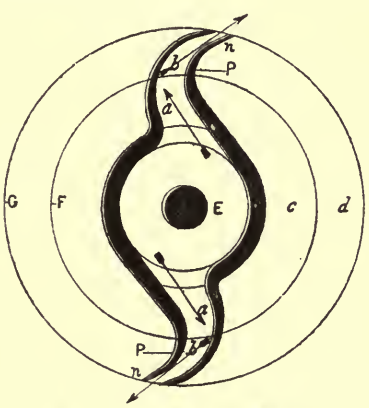


FIG. 39.—Wheel with Narrow-Mouthed Nozzle.

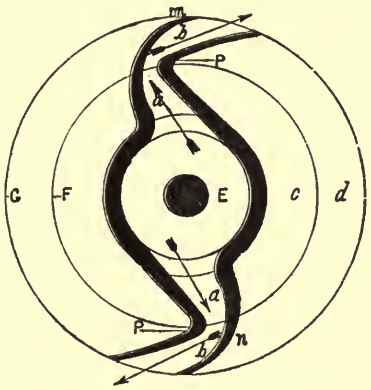


FIG. 40.—Wheel with Wide-Mouthed Nozzle.

In the first case the water fills the nozzle at full pressure, and it escapes at full velocity, the centre of back pressure being at b.

In the second case the water enters the cut-off nozzle at n at full velocity without pressure, glides along the curve, and falls off with little or no velocity. The centre of back pressure is half-way between a and b.

In this Hero form of turbine, then, we see that a very simple alteration in the guide and exit passages makes all the difference between a pressure and an impulse turbine, and that the guide blades are not necessarily fixtures.

Now again, if we take the form (Fig. 40) in which the water issues from the nozzle

at full velocity, and mount another wheel free to move in the opposite direction to the reaction wheel inside, as shown in Fig. 41, both wheels would revolve in opposite directions, the inner being a pressure turbine and the outer an impulse turbine.

FIG. 41.—Combined Pressure and Impulse Turbine.

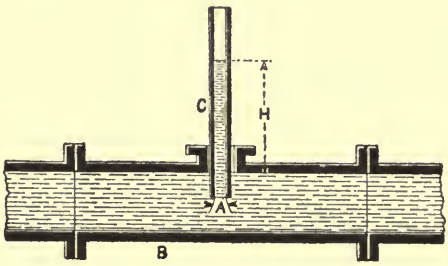


FIG. 42.—Hydraulic Experiment.

In this turbine the two opposite revolving wheels are geared to one horizontal shaft by 3 bevel wheels.

We are now prepared to consider the various types as constructed, but first we shall note the usual forms and facts regarding water and the hydraulic formulæ.

Atmospheric pressure is roughly estimated at 14.7 lbs. per square inch, with a 30-inch barometric height. And the column of water this can support in a water barometer is 34 feet.

Let H represent head of water, in this case the head is 34 feet; hence $\frac{34}{14.7} = 2.3$ feet head = 1 lb. pressure per square inch; or, pressure $= \frac{14.7}{34} = 0.432$ lbs. per foot-head.

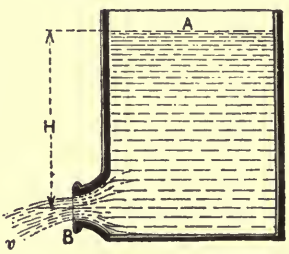


FIG. 43.—Hydraulic Experiment.

A cubic foot of water weighs 62.5 lbs. if fresh, and 64 lbs. if sea water.

Let P = pressure per square foot, H height of column, A its area in square feet, and W the weight of water per cubic foot; then $PA = WHA$, and $P = WH$, and $H = \frac{P}{W}$. These show the connection between head and pressure. Thus if we connect a long vertical pipe C to a water supply main B the water will rise to a height H (Fig. 42), due to pressure P.

Now, as to velocity, if water issues from a converging nozzle (Fig. 44) on a vessel

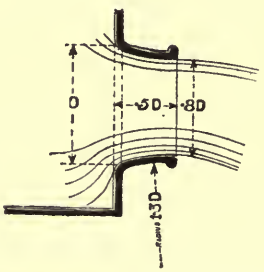


FIG. 44.—Hydraulic Experiment.

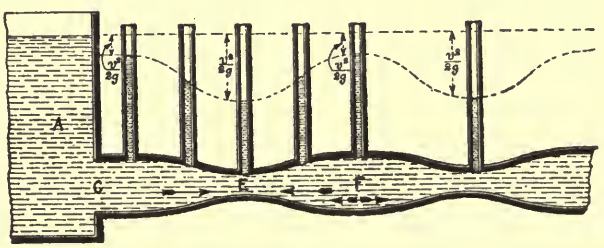


FIG. 45.—Hydraulic Experiment.

as in Fig. 43, we can imagine a single drop of water of weight W falling from the surface to the orifice. When on the top at rest it has potential energy WH, but on falling it has acquired kinetic energy $= \frac{WV^2}{2g}$, so that $WH = \frac{WV^2}{2g}$, and $V = \sqrt{2gH}$; or approximately $= 8\sqrt{H}$.

The conversion of head pressure and velocity is shown very well by Froude's experiments shown in Fig. 45. If a pipe of varying bore is attached to a water vessel, and small vertical pipes are attached to the discharge pipe at the different bores, the water will rise in these small pipes and indicate the side pressure at the different points. As the same quantity of water passes each section of the pipe the velocity of the flow is inversely as the section, and the pressure inversely as the velocity; hence the indicator pipes show the difference in pressure by the height of water. If this difference of height between the level of the water in the tank and the pipes is measured it will be found to be equal in each case to $H = \frac{V^2}{2g}$, where H is the difference between the pipe levels and tank level.

Thomson's water jet pump (Fig. 46) is sometimes explained on the hypothesis that the high pressure jet acquires such a velocity on discharging at the nozzle that the pressure falls below that of the atmosphere, but the experiments of Siemens on steam

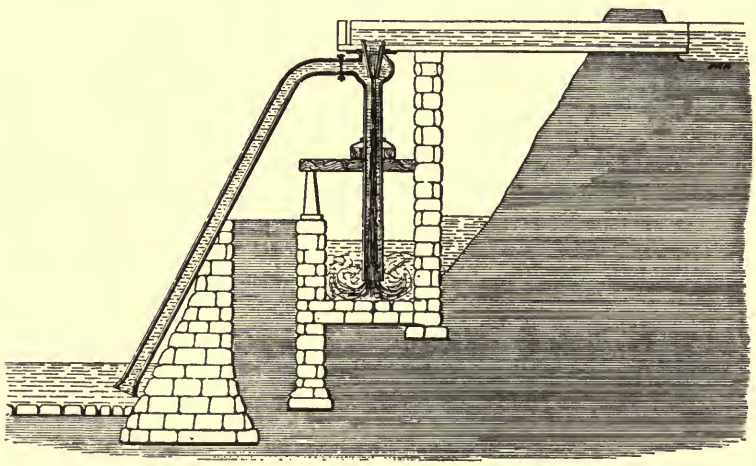


Fig. 46.—Thomson's Water Jet Pump.

jet pumps and others prove that friction between the high velocity jet and the air carries off the air with the jet, and thus produces a partial vacuum behind the nozzle. This raises the water until the two streams meet, one at high velocity of small section and the other of large section and small velocity. The velocity of the one is decreased by expending its energy in increasing the velocity of the other larger stream sufficient to discharge the larger body of water through a pipe widening towards the outlet, at a higher level than that from which it is drawn.

The same action takes place in the locomotive, where the blast pipe carries by its high velocity jet of steam a large stream of air and gas at a slower velocity out against the atmospheric pressure. This jet effect will be further referred to under steam power.

When water flows from a reservoir from a pipe B to a lower level *n*, and discharges through an orifice of very small size compared with the pipe, the whole head is practically applied to the orifice, as the velocity in the pipe may be so small as to be neglected; but if the discharge is comparatively large (Fig. 47), or the orifice area a large fraction of the pipe section, then the whole head is expended in the three forms of H due to head of unexpended fall—$\frac{P}{W}$ due to pressure, and $\frac{V^2}{2g}$ due to velocity, and multiplying each by W gives the energy in each form; and in 1 lb. of water the total would be $E = H + \frac{P}{W} + \frac{V^2}{2g}$ at the discharge.

The turbines to be considered act either by one or two or all three of these forms of energy. In the impulse turbine and jet turbine, like the Girard and Pelton wheel, we want the full velocity at the jet; hence the fall pipe must be large enough in area to reduce the velocity in the pipe to a negligible quantity, so that the full pressure due to head is at the orifice of the jet, and not expended in the pipe.

In the pressure turbine we require the water delivered to the turbine under pressure, so that as the water passes through the wheel it losses its momentum in a forward direction. If V is the forward component of the entering velocity, each lb. of water changes its momentum $=\frac{V}{g}$; hence $\frac{V}{g}$ gives the forward pressure on the wheel due to each lb. of water per second. If V_1 is the velocity of the wheel at the entrance of the water, the work per lb. of water will be $=\frac{VV_1}{g}$ foot-lbs. per second.

And as the energy given by 1 lb. of water at head H is H foot-lbs., of which a large fraction is given to the wheel, $\therefore H=\frac{VV_1}{g}$, the energy equation for turbines.

The guide blades are curved to give a large forward velocity. Practically, the velocity in pressure turbines at the entrance to the wheel is half that due to the head, and the pressure half that due to the head.

The quantity of water theoretically discharged by an orifice is the product of velocity V and the area of the orifice A. Cubic feet per second $=Q=AV$, and $V=8\sqrt{H}$ $\therefore Q=8A\sqrt{H}$. But owing to friction this, in practice, is reduced by 0.96.

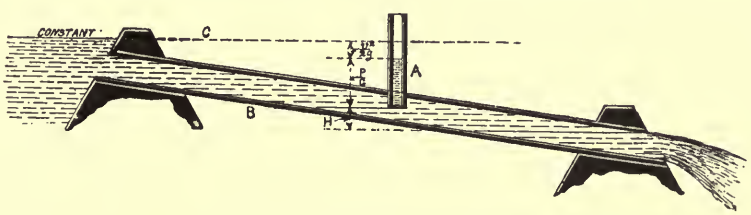


FIG. 47.—Hydraulic Experiment.

In practice we take the effective head at the wheel by a pressure gauge, as no coefficient can be given for loss due to pipe resistances. Again, we need not trouble about the discharge through thin plates or peculiar tubes in this place. In all turbines the orifices are shaped as near as possible to the form of a converging nozzle, so that the area of the orifice is equal to the area of the water jet. Allowances must also be made for friction and bends in pipes, all of which are rather beyond our present scope.

Momentum of water, as we have seen above, is the important factor in turbine power. In theoretical mechanics the unit of force is 1 lb., and of mass 32.2 lbs. $=g$, or mass $=\frac{W}{g}$.

Velocity V is feet per second. $Vt=$ feet $\times$ seconds. $\frac{V}{t}=f=$ acceleration which is the increase of V in each second. If any force is constant the acceleration is uniform. This force being measured by the increase of momentum it produces. Momentum $=$ mass $\times$ velocity, and hence force producing acceleration $=\frac{W}{g}f$; also $Pt=\frac{WV}{g}$, and if P is pressure and t time $=1$ second, W weight of water flowing per second, $\frac{WV}{g}=$ change of momentum, and $P=\frac{WV}{g}=$ change of momentum.

To apply these formulæ to turbines we must take only useful cases.

1st case, the Pelton wheel bucket or cup. The velocity of the jet is V_1, and of the bucket forward is V_2. When the jet strikes the relative velocity is $V_1 - V_2$ forward, and when the jet is turned back $V_1 - V_2$ backward.

The absolute velocity of the jet is V_1 before it strikes, and after $V_2 - (V_1 - V_2) = 2V_2 - V_1$ weight of water per second is $WA(V_1 - V_2)$; therefore the pressure on the cup = difference of momentum = $\frac{WA(V_1 - V_2)V_1}{g} = \frac{WA(V_1 - V_2)(2V_2 - V_1)}{g} = 2\frac{W}{g}A(V_1 - V_2)^2$.

From this it is easy to deduce that if $V_2 = \frac{1}{2}V_1$ the absolute velocity of rejected water is 0, and all the energy of the water given up to the wheel.

In all cases impulse pressure on the turbine blades is

= momentum before entering – momentum on exit.

In pressure turbines it is = (momentum + pressure) – (momentum + pressure) before and after passage through the wheel.

In the Hero reaction wheel momentum is of no account whatever, unless the outlets are curved to form impulse wheel blades; then in that case we get the back pressure $= pA +$ the momentum of the fluid entering the curved exit – the momentum on leaving the curved exit. But as it is difficult to shape the orifice so as to change the momentum entirely in its passage it is necessary, in using reaction wheels, to compound them with an ordinary impulse wheel, so that the remaining energy in the jet is absorbed by this additional wheel. This is more important with wheels driven by steam, air, or gas pressure, wherein the fluid velocity is very high, and we shall deal with this under steam turbines.

In the practical construction of turbines all these principles are applied, and upon them the dimensions of the parts of the turbine are calculated for the best effects.

We shall now consider the practical turbines of present day manufacture. The simple Pelton wheel has proved the best for high falls under most circumstances, and we may begin with that turbine. It is not to be supposed that it was designed by its inventor on scientific knowledge that the pressure was greatest on the cups when $V_2 = \frac{1}{2}V_1$, or that completely turning back the jet gave twice the pressure, which he obtained by striking the jet on the old flat board vanes of the primitive wheel,—the miner's "hurdy-gurdy wheel,"—which was built in ignorance of all hydraulic laws, yet it did good work, and probably had an efficiency of 40 per cent. Its great advantage, however, was that it could be made on the spot from the timber at hand, rough and readily put up.

An interesting paper, read before the American Institute of Mining Engineers by W. A. Doble, may be consulted on this wheel, from which the following details are obtained.

It seems that the double cup with the mid-rib was invented by several engineers, but Mr. Pelton certainly has the credit of introducing it.

Mr. Pelton's own account of his invention is interesting. It appears that one of the cup-shaped wheels already mentioned got loose on its shaft, and, being displaced relatively to the jet, the latter struck the side instead of the centre of the bucket. It was observed that under these conditions the mill which the wheel was driving ran faster. As, however, the eccentric position of the jet caused an end thrust on the shaft, Mr. Pelton proposed first to have two sets of buckets mounted on either side of the wheel, the side thrust on the one set of buckets being counterpoised by that on the other, the jet playing on the dividing line between the two sets of buckets, as indicated in Fig. 48, for which we are indebted to Mr. Doble's paper. The step from this arrangement to the splitting wedge was obvious, and was made by Mr. Pelton, a mid-section through whose bucket is shown in Fig. 50, whilst Fig. 49 shows a complete bucket in perspective. This type of bucket soon came into general use, and with it the claim for very high running efficiencies. How far these claims are sub-

FIG. 48.—Pelton Wheel.

stantiated is even yet a matter of dispute, the results obtained in different careful experiments varying through very wide limits. The highest figure which seems at all well authenticated is one of 91.85 per cent., which is said to have been attained in the case of a 26-inch cascade wheel tested by Professor Hitchcock in the engineering laboratory of the Ohio State University in 1897. This figure was obtained with a head of 163 feet; the flow of water was 2085 lbs. per minute, and the bucket velocity 46.2 feet per second. The wheel ran at 447 revolutions per minute, and gave 35 horse-power on the brake. A similar wheel of larger size, giving 80 brake horse-power, showed an efficiency of 90.04 per cent. In the absence of complete details as to the method of conducting the test these figures will be treated with some reserve, and perhaps the same should be said as to certain tests made by Mr. Ross E. Browne in the laboratories of the University of California in the year 1890. Here a Pelton wheel 15 inches in diameter, working with a $\frac{7}{16}$-inch nozzle under a head of 50 feet, is stated to have shown an efficiency of 82.6 per cent., whilst with a $\frac{3}{8}$-inch nozzle the efficiency was 82.5 per cent. On reducing the head to 8 feet the efficiency was 73 per cent. On the other hand, in some experiments made in 1897 at the M'Gill University, by Mr. J. T. Farmer, the wheel made a much less favourable showing, the highest efficiency recorded being 70 per cent. The wheel tested in this case was a Pelton No. 3 motor, 18 inches in diameter. The experiments were made with heads up to 235 feet, and with nozzles

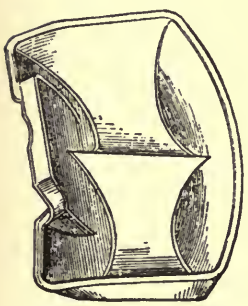


FIG. 49.—Pelton's Bucket.

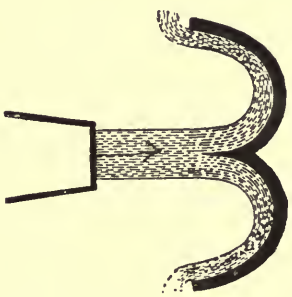


FIG. 50.—Section of Pelton's Bucket.

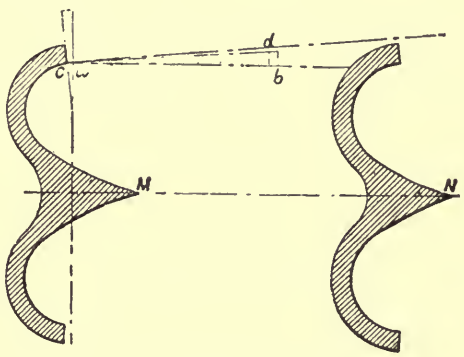


FIG. 51.—Section of Berry's Bucket.

$\frac{1}{2}$-inch and $\frac{3}{4}$-inch in diameter, whilst the speed was varied both above and below the point of maximum efficiency, which, as stated, was but 70 per cent. This, no doubt, is a very fair result, but is far below those claimed in the tests made by Professor Hitchcock and Mr. Browne. If the figures obtained by these observers can be accepted, it would seem that there is practically no margin left for any substantial improvement in the tangential water wheel. Nevertheless, as commonly constructed, there are several possible causes of dissipation of energy. In the first place, the angle at which the jet strikes the bucket varies as the wheel turns round, whereas in a well constructed impulse turbine this angle is constant, and is, moreover, more nearly tangential to the direction of the jet than is the case with the bucket of a tangential wheel.

Further, as often constructed, the flat lower lip of the bucket slaps the jet instead of entering it quietly. This defect has probably arisen from the makers neglecting to take into account the relative motion of the bucket and of the jet. A type of bucket patented by Mr. S. L. Berry, Philadelphia, has no lower lip at all, but is brought to a sharp edge. The back of this entering lip is bevelled away so that it will stand clear of the jet throughout its motion. The jet-splitting wedge is curved in the vertical plane. This curve is chosen so that the vertical angle between the jet and the wedge changes little throughout the action of the jet. This curve is easily drawn by determining the virtual direction of the jet at three points in the path of

the bucket and drawing a circle through these. As shown in Fig. 51, the jet on leaving the wheel does so in the direction of *cd*, thus clearing the next bucket; it carries away with it, therefore, residual energy due to the transverse velocity *bd*. It will be seen that in order that this should be small it is necessary that the buckets of a tangential water wheel should not be too closely spaced, as otherwise the jet on leaving one bucket will strike on the back of the next. The bucket just described would seem to have several points of superiority as compared with the ordinary Pelton bucket. The latter in another way also violates hydrodynamic principles, since it is a maxim in hydraulics to avoid any kind of sharp corner or angle to which the flowing water may have access. In Fig. 49 it will be seen that there are a number of sharp corners in the bucket. It is true that the bulk of the water will avoid these, but since a jet striking a surface at an angle spreads in all directions, some water must find its way into these corners, and some loss by eddying must thereby arise. A bucket which appears to have special merits in this regard is the Doble ellipsoidal bucket, which is illustrated in Figs. 52 to 55. In this bucket the lower edge is, it will be seen, cut away near its centre, and the jet-splitting wedge E terminates in a sharp point F, which will enter the jet as the bucket comes round with a minimum of disturbance. Whatever direction the water may flow in, it meets a surface of easy curvature, and is delivered over the edges of the bucket with a proper residual velocity. We have no particulars as to the absolute efficiency attained with wheels thus fitted, but they have been used to replace other buckets on the wheels of the San Joaquin Electric Company, with an increase, it is stated, of 12 per cent. in efficiency. This statement may, of course, mean much or little, dependent upon the efficiency of the wheels in their original state; but theoretical considerations, as already mentioned, would lead us to anticipate that the Doble buckets should prove exceedingly good whenever they are submitted to a properly conducted test. Another feature claimed for them is that they wear very uniformly when the water supply is charged with sand. Thus in the Mount Whitney Power Company's plant, although the working head is 1300 feet, and no sand traps were originally fitted, the wheel was still in excellent condition at the end of fifteen months.

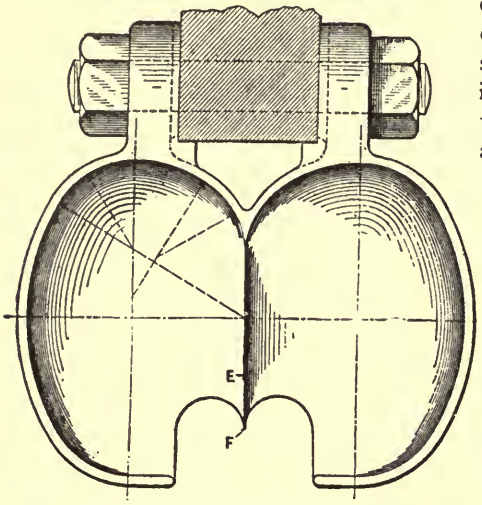


FIG. 52.—Doble's Bucket. Plan.

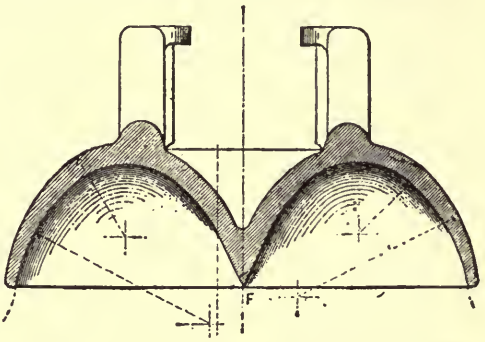


FIG. 53.—Doble's Bucket. Sectional Elevation.

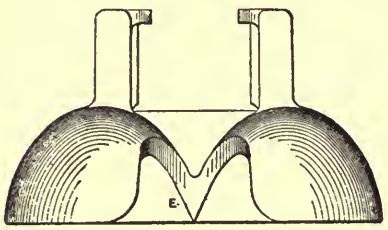


FIG. 54.—Doble's Bucket. Elevation.

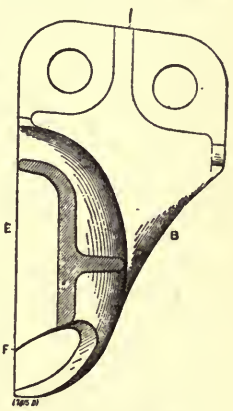


FIG. 55.—Doble's Bucket. End View.

After this a sand trap was brought into use, and it was found that the water, though clear, was highly charged with granite sand. A pit 100 feet long by 6 feet wide fills up 4 feet in from three to five days, so that during its 15 months' work the buckets must have been subjected to the scour of 500 or 600 cubic feet of sand per day. As the velocity of the jet is about 295 feet per second it is only natural that the buckets were somewhat worn; but this wear has been uniformly distributed, and the surfaces are as smooth and perfect as when originally fitted. One of these turbines is illustrated in Fig. 56. The wheels are 4 feet in mean diameter, and are designed to run at 250 revolutions per minute. The most usual plan of governing wheels of the Pelton kind has in the past been that of simply deflecting the jet, so that it failed to strike the buckets. This plan is, of course, wasteful of water, since as much is sent into the tail race when the wheel is running light as when loaded, and in practice is less simple to carry out than might at first sight appear. In short, the nozzle is mounted on a ball-and-socket joint, such as is used with hydraulic mining monitors. This joint has to be kept tight under high pressures, and yet it must nevertheless move easily when necessary, if "hunting" is to be avoided and the speed of the wheel kept fairly constant. The difficulties in effecting this led to the very ingenious system of governing by Cassel.

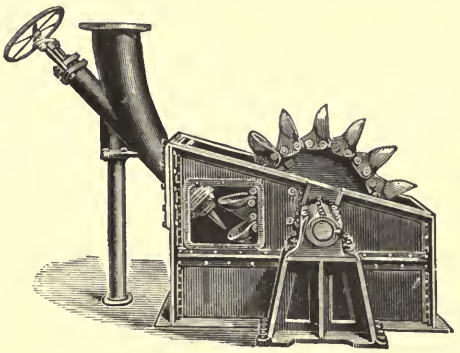


FIG. 56.—Type of Pelton Wheel.

In this case (Fig. 57) the wheel is built up of two discs mounted on the same shaft, along which they are free to slide, but are normally held in contact with each other by springs. Each disc carries a series of half Pelton buckets as shown. So long as the two half buckets are in contact the wheel acts as an ordinary Pelton wheel. The discs, however, are provided with weighted levers so arranged that under the action of the centrifugal forces developed on the rotation of the wheel the two discs are forced apart as indicated in Fig. 57, so soon as the speed rises above a certain predetermined limit. The jet then passes between the wheels and is wasted in the tail race. With this arrangement the wheel is its own governor, and accordingly the control of speed is remarkably prompt and efficient.

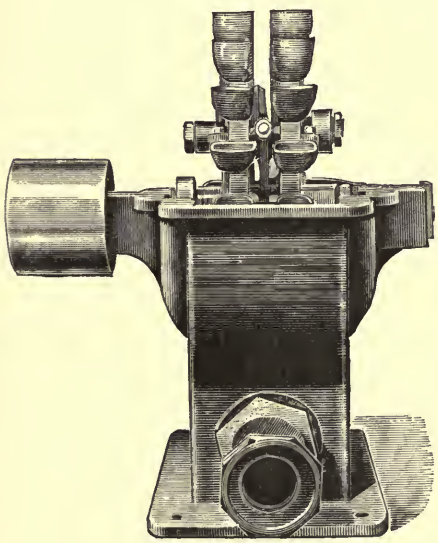


FIG. 57.—Type of Pelton Wheel.

The plan of reducing the pressure producing flow through the nozzle by means of a throttle valve has also been tried, but is objectionable, in that the speed of the jet is thereby reduced and the water no longer enters the wheel without shock, a waste of energy thus arising. A simple plan, which we can well believe gives very good results, is described and illustrated in Fig. 58. In this case a sharp edged sluice is, it will be seen, fitted before each nozzle, and the wheel is governed by moving this sluice so as to cover more or less of the opening. This method affects the velocity of the jet very little, but changes its shape, and there may therefore be some loss from splashing, owing to the inversion of the jet. As is

well known, to be stable a jet must be circular in section. If of any other form the surface tension tends to bring it to this form, but overdoes the correction, so that in point of fact the shape of the jet oscillates about the circular section. It is quite probable that with heads of fair weight the velocity of the jet will be too great for any action of this kind to have any perceptible effect in the passage between the jet and the buckets. Mr. Doble's arrangement is of the throttling type, but the details of the construction are such that a solid cylindrical jet is obtained, issuing at practically constant velocity at all gates, from full bore down to under one-fifth full bore. The throttle piece is pear shaped, and is maintained central with the jet by being mounted on a cylindrical rod passing through the back of the casing, and which can be moved in or out. The water issues through the annular space between the throttle piece and the jet. The outer end of this throttle piece projects always through the orifice of the jet, and terminates in a fine point connected by easy curves with the thickest portion of the body. The water clings to these curves, so that the jet is always cylindrical and solid. Figs. 60 and 61 show the jet obtained with various openings of the nozzle. The steadiness and transparency of the jet is remarkable (Fig. 59).

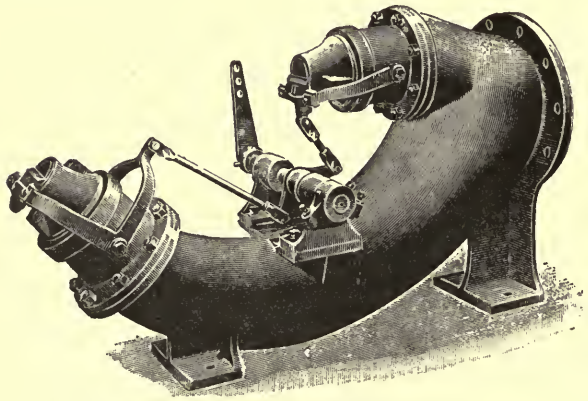


FIG. 58.—Sluice Governors of Pelton Wheel.

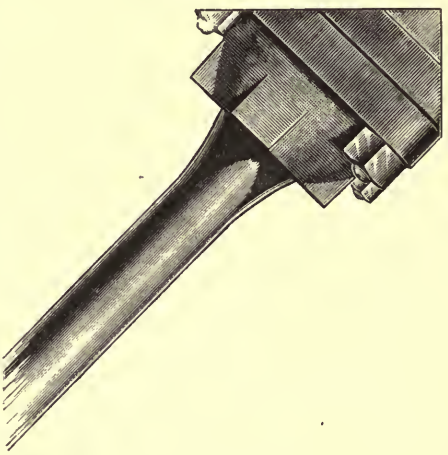


FIG. 59.—Jet from Doble's Nozzle.

Nearly all tangential wheels act much in the same way as this Pelton wheel. The Girard turbines being much the same in principle, although very different in construction, in both the momentum of the jets is converted into pressure on the wheel cups or blades in being arrested and turned back in its motion; and while the angular velocity of the wheel at the wheel blades is necessarily about half the linear velocity of the jet, the revolutions of the wheel may be chosen and arranged for by calculating the diameter necessary to obtain the angular velocity at the given speed, and from this the radius, or

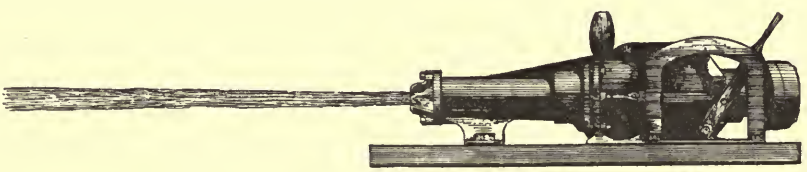


FIG. 60.—Jet from Doble's Nozzle. Full open.

one of the radii r of the wheel, may always be found; thus in a wheel like Pelton's r is the radius from centre of shaft to centre of cups. Suppose that half the velocity of the jet is 30 feet per second, and the wheel limited to 10 revolutions per second, then

$\frac{30 \text{ ft.}}{10 \text{ revs.}} = 3$ feet as the circumference at radius r, the centre of the buckets from which r is found.

In the outward radial flow (Girard) there are the inner radius r and the outer r_2; and having found r, r_2 is found by $k \times r$, wherein k is a constant about = 1.17 to 1.18.

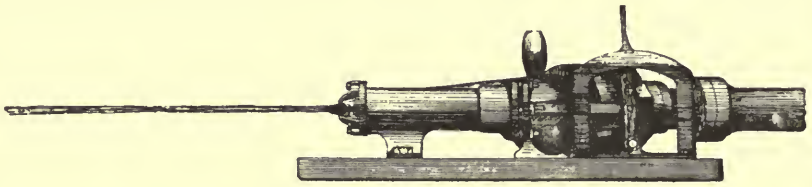


FIG. 61.—Jet from Doble's Nozzle. Partially open.

THE GIRARD TURBINE

This is an impulse turbine in which the wheel buckets are not under pressure, but may be a full or partial flow wheel.

It is made in various forms, like all other turbines. Thus for high falls it can be placed at the bottom of a wheel pit and the water led down by a pipe, so that it enters either between twin wheels or under one wheel, and supports the weight of the wheel and shaft, as in the case of Niagara Falls.

And, again, for high falls where slow rotation is desired it can be made a partial flow wheel, so that although the rotations are low the peripheral speed is still high enough for good efficiency.

Fig. 62 represents Messrs. Gunther's turbine of the class for deep wheel pits. When a turbine is placed in a deep pit, and the upright shaft is of extreme length and of great weight, it may be desirable to reduce the load on the footstep as much as possible, and in such cases a Girard turbine of special design with radial outflow is adopted. In this type the water enters the turbine from the underside, is admitted on the inner circumference of the wheel, and is discharged radially outwards; the water pressure thus exerts thrust on the wheel, and the footstep has only to carry the weight of the moving parts.

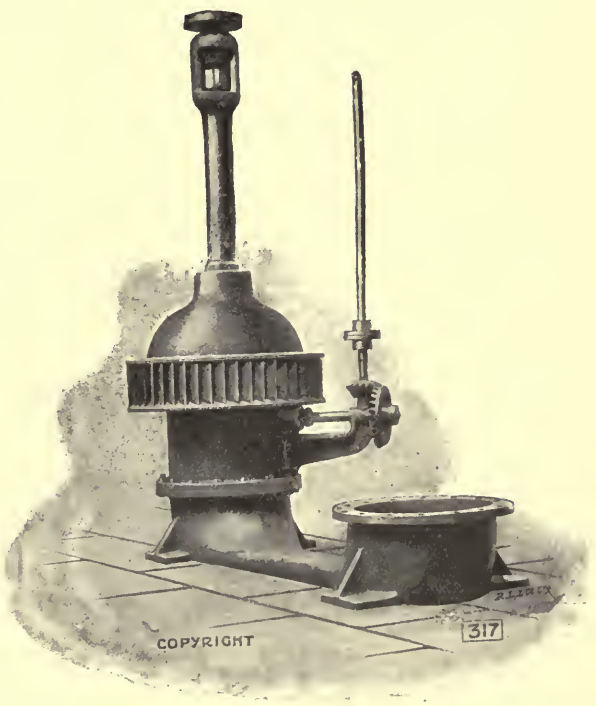


FIG. 62.—Girard Turbine for Deep Wheel Pits.

Fig. 63 is from a photo of one of two radial Girard turbines driving a large textile mill abroad, each wheel being 63 inches internal diameter, and of 525 horse-power, with 95 feet fall; the two turbines drive by steel bevel wheels on to the same second motion shaft, and each upright shaft is about 80 feet long.

The two diagrams, Figs. 64 and 65, show the construction of the partial flow turbine of this type. G is the inlet nozzle, furnished at its nose with two or more guide blades movable by rack and pinion for regulating purposes. It will be seen from this that the wheel's diameter is independent of the velocity of water flow.

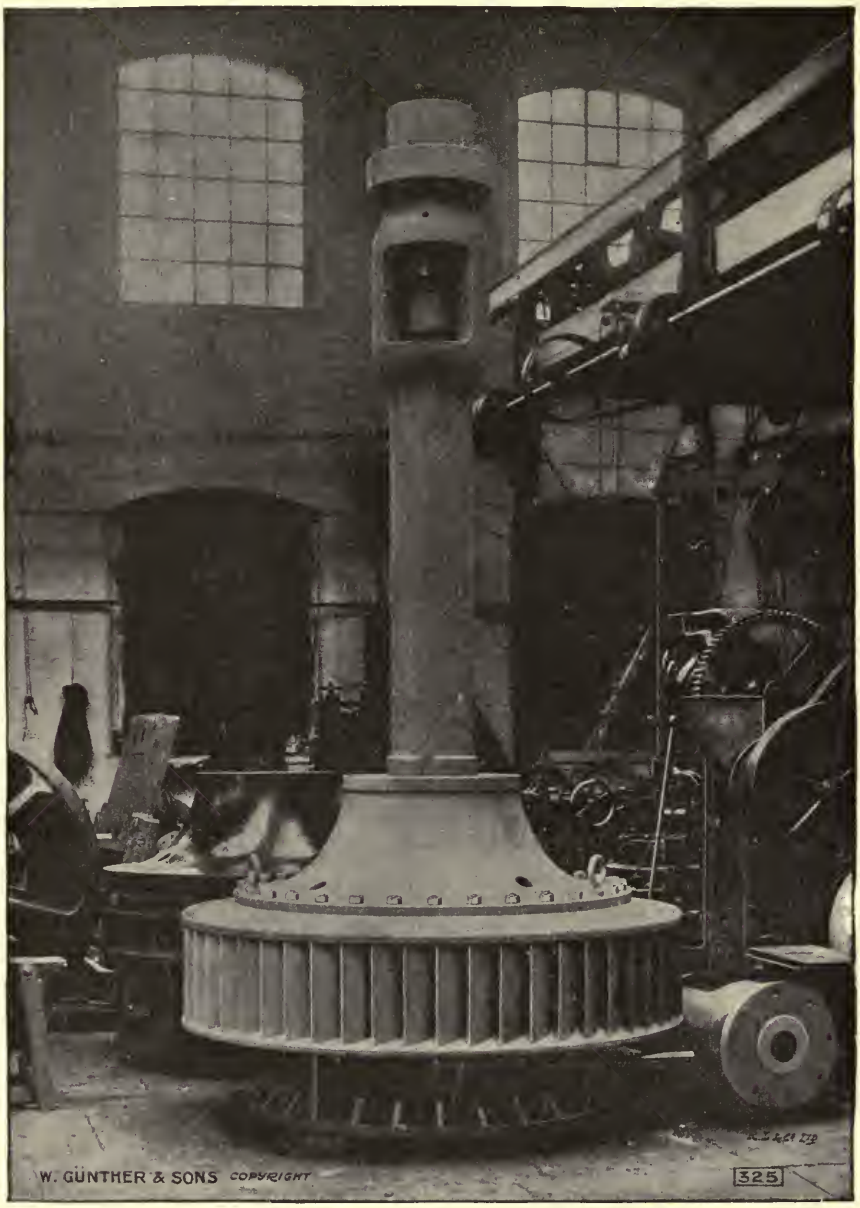


FIG. 63.—Girard Turbine with Radial Outflow.

For this class of turbine we will take a partial flow example with a high fall, and not a great quantity of water. Let us take—

h as the head in feet = 400.

Q the quantity of water available in cubic feet = 4 per second.

R = 500 the limit of revolutions of the shaft per minute.

A is the area of the guide passages.
A_2 the area of the wheel passages at entrance or inner radius.
r is the outer radius wheel.
r_2 the inner radius of the wheel.
C the velocity of the inner radius in feet per second.
C_2 velocity of the outer radius in feet per second.
V the velocity of water issuing from guide passages.
V_2 velocity of water from wheel.

Then $V = 0.94 \times 8\sqrt{h}$ (1).

$$C = \frac{0.94 \times 8\sqrt{h}}{2} \quad (2).$$

$$A = \frac{Q}{V}.$$

$A_2 = \frac{Q}{V} \times 1.08$, as the wheel blades must never be filled.

$r_2 = C$ (found by taking the revolutions per second) $\div$ R (in this case $\frac{506}{60} = 8.3$), —and dividing C by this number $= \frac{C}{R}$, the product will give the circumference of the inner r_2, from which r can be found.

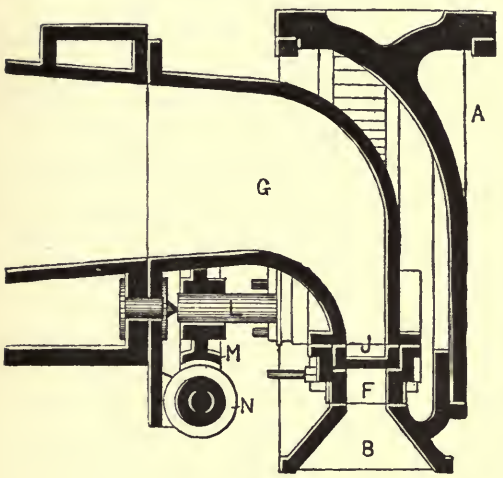


FIG. 64.—Section of Girard Turbine.

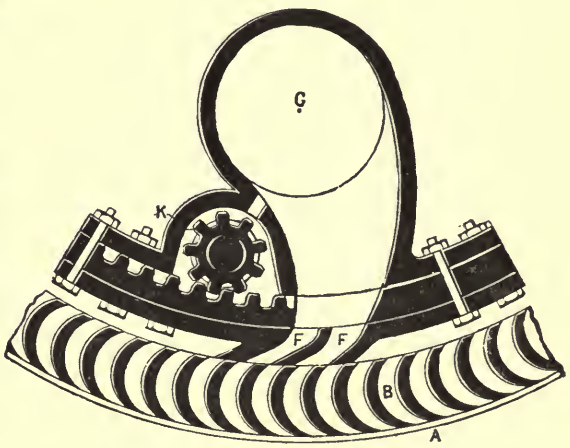


FIG. 65.—Section of Girard Turbine.

Beginning with C, we find that $C = \frac{0.94 \times 8\sqrt{h}}{2} = \frac{0.94 \times 8\sqrt{400}}{2} = \frac{0.94 \times 8 \times 20}{2}$ $= \frac{150}{2}$ feet $= 75$ feet per second; and the value of $r_2 = \frac{C}{R} = \frac{75}{8.3} = 9$ feet as circumference of r_2, from which we get easily 1.5 feet as the radius of r_2.

The area of the guide blade passages is $= A = \frac{Q}{V} = \frac{4}{150} = 0.026\dot{6}$ square feet, or nearly $3\frac{3}{4}$ square inches.

Area of wheel passages $A_2 = A \times 1.08 = 3.75 \times 1.08 = 4$ square inches.

The circumference 9 feet, found above, would be the inner circumference, and for the purpose of calculating the axial breadth of the wheel blades we must find the number of blades required.

The guide blade passages $A = 3.75$ square inches; and if we divide this between three orifices in the nose ot the guide each would be $\frac{3.75}{3} = 1.25$ square inches area, say $1\frac{1}{4} \times 1$; $1\frac{1}{4}$ inches axially and 1 inch measured circumferentially.

The wheel blades passages are somewhat larger = 4 inches, and axially they would be the same as the guide blades, $1\frac{1}{4}$ inches; and if we divide the 4 inches into 3 we get $\frac{4}{3} = 1.\dot{3}\dot{3}$ inches as the width between the blades on the wheel at the inner radius r in inches; we must add to this the thickness of the blades. In a wheel of this high velocity for a small quantity of water it would be made as light in weight as possible and well balanced. The blades would be no more than $\frac{1}{8}$ inch thick, hence from centre of one blade to centre of next would be $= 1.\dot{3}\dot{3} + 0.125 = 1.455$ inches; and this into 108 inches, the circumference $= \frac{108}{1.455} = 74$ blades.

If the guide nozzle had only one orifice of 3.75 square inches we could alter the wheel blade design considerably; thus the wheel orifice, being 4 inches, could be divided into two of 2×2 inches, and the nozzle, one orifice, $\frac{3.75}{2} = 1.875$ inches and 2 inches axially measured. This would give 2.125 from centre of one blade to the next on the circumference; hence $\frac{108}{2.115} = 50$ blades.

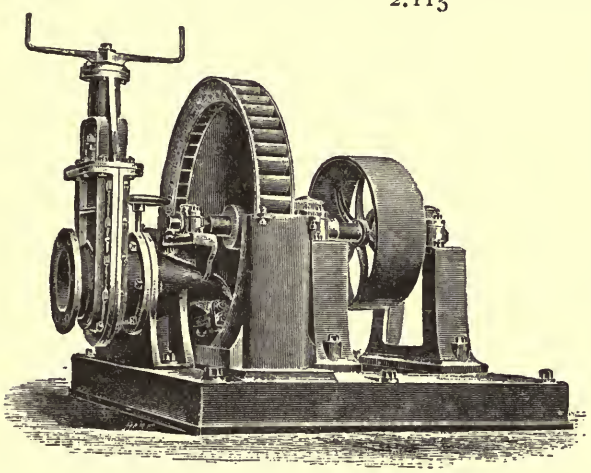


Fig. 66.—Girard Turbine Complete.

Fig. 64 illustrates the sections of the wheel and guide blades and passages, and Fig. 65 the orifices, nozzles, methods of regulation by a sliding sluice worked by rack and pinion.

The blades are splayed out, as formerly explained, to enable the water to spread as it passes through the passages, so that the outer area of the passages is somewhat greater than their inner passages.

The outer radius r should be to r_2 as 1.18 is to 1. In this case r_2 is 18 inches; hence $r = 1.18 \times 18 = 21\frac{1}{4}$, and the buckets would therefore be $3\frac{1}{4}$ inches radially.

The efficiency is about 75 per cent. at best, so that as the power $= \frac{Q \times h \times 62.5}{33{,}000}$, $Q = 240$ cubic feet per minute, and $h = 400$, 62.5 the weight of one cubic foot, hence $0.75 \frac{240 \times 400 \times 62.5}{33{,}000} =$ about 130 actual horse-power.

There is no pressure between the guide blades and the wheel, and as the water enters the buckets without pressure it is freely deviated by them, and takes a course quite independent of their shape. The action of the water on the wheel depends on the angle through which each particle is deviated whilst freely flowing over the buckets, and as these latter are not full there is no disturbance of the action as they pass in front of, or away from, the jets.

The turbine illustrated in Fig. 66 is shown with part of the hood removed, which prevents the water from flying about. The water enters through the valve shown at the left-hand side of the Figure, and passes directly into a distributing chamber, from which it jets on to the buckets of the wheel through gates or ports, varying in number according to circumstances. When there is only one port it can be reduced in size when water is scarce or less power required; and when, as is frequently the case, there are many ports, a sufficient number of them may be used to suit the requirements of the moment. The arrangement by which the power is regulated is easily worked either by hand or by a governor.

THE VORTEX WHEEL

These are inward flow turbines with fixed guide blades, and include the Thomson vortex wheel with a few movable guide blades for regulating the quantity of water and power.

This vortex turbine has much to recommend it. The regulation is easy and certain, due to the balancing of the inward pressure by the outward pressure, due to centrifugal force. In fact, it will interest electricians to study this turbine, as it is a hydraulic illustration of the electromotor, and both act upon the same fundamental mechanical law. This turbine, when supplied with water under pressure, acts as a motor and gives off power, but at the same time it acts as a centrifugal pump, and hence produces a counter pressure, and it works at its greatest activity when the counter pressure is half the working pressure. A pump is a water pressure generator, a turbine is a water pressure motor, a dynamo is an electric pressure generator, an electric motor is an electric pressure motor. The vortex and other inward radial flow turbines act in both capacities when at work. In the electromotor a counter pressure is generated in the same way, and the greatest activity is also obtained when the counter pressure is half the working pressure, but its efficiency is greater the nearer the counter pressure approaches the value of the working pressure.

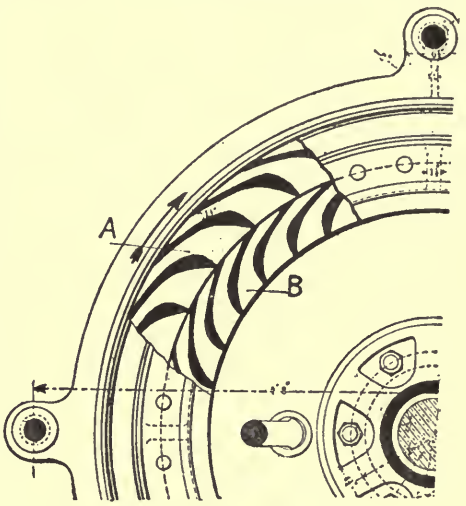


FIG. 67.—Turbine at Niagara Falls.

In considering this fact the author advocates the inward flow radial type of turbine for single expansion steam turbines, in which the counter pressure would moderate the steam flow. However, that is for another chapter.

We shall enter fully into the consideration of this type of turbine, for with this, the Pelton wheel, and improved Jonvals all cases can be met.

The turbine selected for illustration is the Gilke's vortex, and after fully describing it we shall calculate its dimensions. When these are correct and the speed adjusted to the number of revolutions necessary to give the counter pressure about half the working pressure, it runs with great smoothness and silently.

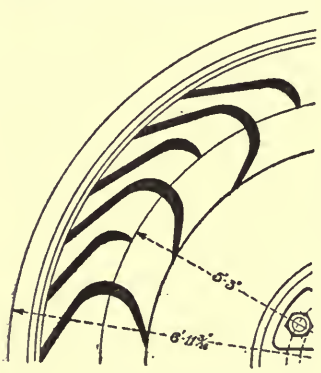


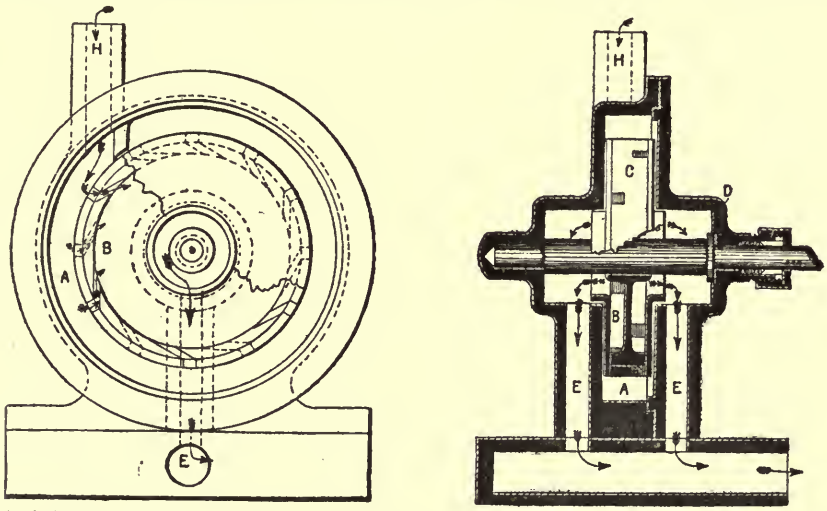
FIG. 68.—Turbine with Rotating Guide Blades.

Referring to the fundamental turbine, the Hero reaction wheel, this vortex wheel can be traced back to that also; for it would work even if the guide blades were fixed to the wheel blades and rotated with them; in fact, it works well so made. This is shown in Fig. 68.

In fact, all pressure turbines can be traced to this simple form of wheel, and by starting from that ancestor we could treat them as reaction wheels, from the size of the orifices at the outlet, and the pressure of water in the wheel, and the centre of pressure.

Let us for a time depart from the orthodox treatment of pressure turbines, and consider them from another point of view. Turbines have been very thoroughly investigated, and the results of practice have been carefully and accurately compiled,

so that new views, while they may not advance the theory much, may, however, have some practical results; indeed, they have already had practical results of value. In further developing the fundamental turbine, suppose we make a wheel as in Fig. 67, the inner curved part of the blades B would, in an inward flow turbine, be fixed, and the inner reverse curves, the wheel, movable. In this case (Fig. 68), however, they are one blade and move together, the pressure water entering on the inner periphery and flowing out at the outer A into the exhaust. We could calculate the power by the pressure difference between the one side of the passage and the other due to the unopposed pressure opposite the exit of the passages, just as we could in a Barker's mill. In this case the inner orifices have a larger area than the outer ones, otherwise there would be no pressure in the passages; we make it an outward flow wheel by a slight alteration, wherein the orifices on the outer periphery are contracted, and the inner ones opened out to admit the water pressure. Here we have the genesis of the outer and inward flow pressure turbines called reaction turbines. First, it is observable that the inward flow has the advantage, in that it is



End View, with Front Cover Removed. Sectional Elevation.

FIG. 69.—Inward Flow Turbine.

natural that the inner orifices should be the smaller on the small circumference, and the large orifices on the outer radius freely admit the water.

Let us see what teaching there is in this theory. First, to revert back to the reaction wheel in its simple form, to maintain the pressure in this turbine the inlet must be large compared with the outlet nozzles in sectional area, in order that the pressure of the water be maintained in the wheel opposite the exit, so as to get the back driving pressure. The exits at B in the inward flow turbine would then be small compared with the entrances at A, and the full pressure due to the head would be maintained in the wheel passages, and the pressure being high would tend to cause great leakage past the sides of the wheel into B. But that can be overcome mechanically, as in the ordinary vortex wheel. A wheel on this principle is shown in Fig. 69, an end view and sectional elevation. The fluid enters at H and fills chamber A at full pressure; it enters wheel C by wide orifices on the outer radius, and is deviated tangentially and discharged through smaller nozzles or orifices in the inner radius, from which it is exhausted through E. These are diagrammatic figures of actual turbines; the entrances to the wheel passages are curved so as to save friction and eddy currents. Now, it is evident that in this turbine only pressure is at work, and very little of that pressure is due to momentum.

The nearest approach to it in action is the vortex wheel, in which reaction is due to half pressure and half momentum, and by altering this wheel so that its outlets are very small compared with its inlets, it can be made also to work with small momentum and large pressure in the wheel passages.

The chief advantage and only *raison d'être* for the use of guide blades in pressure turbines of reaction type is to convert the pressure of the water into momentum in the guide passages, so that the pressure is small at the clearance between the two sets of blades, which must be of some width, and not subjected to high pressure to cause great leakage. The guide blades restrict the inflow of water, and as it finds little resistance owing to the outer or inner discharge orifices being nearly as wide as the inlets, *i.e.* the guide passages, it rushes through at high velocity across the clearance space into the wheel passages; here it is arrested and its momentum reconverted into pressure, and is discharged in the opposite direction from the orifices. This, then, is the object of fixed guide blades, to reduce the pressure by converting it into velocity, so that the pressure is relieved from the exit of the guide blades and the inlet of the wheel passages, enabling the water to jump from one to the other at high velocity and with little side pressures tending to leakages.

Fig. 67 is a part section of the pressure turbines at Niagara Power Generating station, showing the shape of blades and passages very well, and they are certainly of correct form for fixed and movable blades. But the same turbines could be altered to work without guide blades; this I have shown in Fig. 68.

In both cases the pressure water would enter from the inside for mechanical reasons, and discharges into the atmosphere. In the turbine with fixed blades (Fig. 67) there are three conversions of energy: first, the potential energy of the fall is converted into kinetic energy in the guide blades, and each particle is for an instant arrested in the wheel, where its kinetic energy is converted into pressure, which again is converted into velocity with which the water escapes. Under these conditions the velocity of the wheel blades should be $\frac{V}{2}$, where V is the velocity of the water as it jumps the clearance space,—this in order to use the momentum to best advantage.

In the case of the wheel with no fixed blades there is only one conversion of energy. The potential energy of the water is converted into kinetic energy by the difference of pressure caused by the unopposed pressure opposite the orifice moving the wheel round. Theoretically, the best speed would be that due to the head of water, so that the outer radius should have an angular velocity $= V = 8\sqrt{h}$ nearly in this turbine.

The question between the two is, then, whether the guide blades fixed are better and cheaper than the packing devices required to prevent leakages of water in a wheel without them.

The vortex wheel shows that the packing is no difficulty, and its four large guide blades might, if it were not for their excellent regulating functions, be discarded by curving the wheel blades differently, as shown in Fig. 68.

The reason why in all turbines so much depends upon the angles and curves of the blade is simply due to the velocity of the water passing, and its inertia, and momentum. It must be diverted and have its motion altered in direction gradually, and pass from the guides to the wheel without striking a sudden blow or shock.

From these few observations the student will gather the different views on the theories, and their discussion is interesting. We will now proceed with the discussion of the reaction pressure turbine, the vortex wheel.

In pressure turbines the buckets of the wheel must be completely filled with water (as otherwise there would be no pressure), and consequently the turbine must receive the water all over the circumference in order to act efficiently. If this be not done one of two actions takes place according as the turbine be working submerged or not. If the turbine be submerged the buckets, after leaving the guide blade orifices,

are full of still water, which has to be displaced when they come round to the orifices again. Hence in each revolution a considerable amount of power is wasted in imparting momentum to the mass of water carried round in the wheel. If the turbine be not submerged a considerable quantity of water is used in filling the buckets (which have become empty on leaving the guide blade orifices) before the full pressure at the circumference can be utilised. It is not easy to find a mode of regulating pressure turbines which does not involve shutting the water off part of the circumference (so making them partial admission turbines) or otherwise greatly affecting the conditions of the admission of the water. The vortex turbine is only affected by back water to the extent that the water level is affected. It can be placed above the bottom of the fall, so that part of the head can be utilised by suction. And in many cases it is the cheapest form of turbine to adopt.

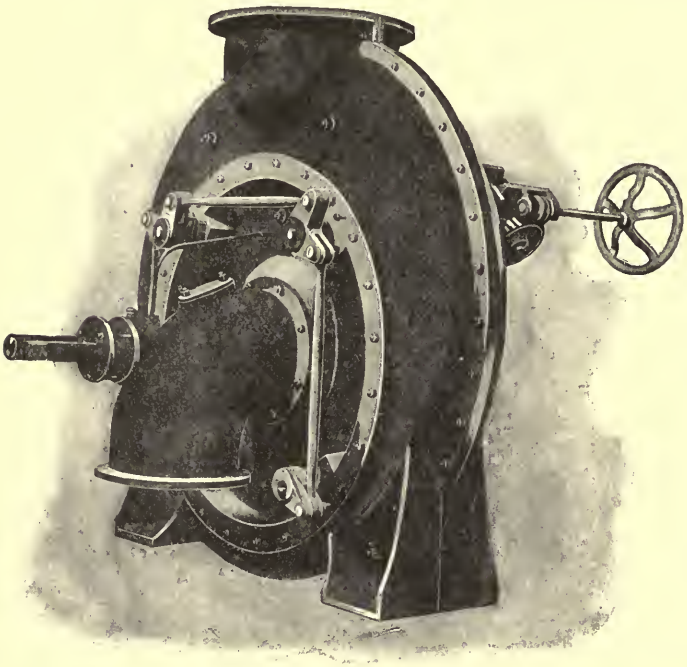


FIG. 70.—Vortex Turbine.

The principles of the vortex and the mode in which the water is applied in it will be readily understood by reference to Figs. 70 and 71.

Fig. 71 shows the internal arrangement of the vortex. The water enters the outside casing at the top,—or in any other position that may be convenient,—and passing thence is directed by four (or more) guide blades on to the outer circumference of the revolving wheel, which is driven round at a velocity depending on the height of the fall. The water, having

FIG. 71.—Vortex Turbine with Cover Removed.

expended its energy in giving motion to the wheel, is discharged through the two central openings, half the amount being carried away by each suction pipe. The guide blades, it will be noticed, are movable, and turn about on a pivot placed near their inner ends. The method in which these guides are simultaneously regulated will be seen by reference to Fig. 70, which is an outside view of the same turbine.

The power is obtained with a slower velocity of water than in ordinary turbines. This is effected by the balancing of the centrifugal force of the water in the revolving wheel against the pressure due to half the head, so that only one-half of the fall or head is employed in giving velocity to the water, the other half acting simply in the condition of fluid pressure. Hence the velocity of the water in no part of its course exceeds that due to one-half of the fall, and the loss from fluid friction and agitation of the water is thus materially less than in other turbines where the water is required to act at much higher velocities.

From the principle of injection of the water from without towards the centre, which is adopted in the vortex, there results another saving of effect, since it admits of the use of long and well formed channels by which the water is made gradually and regularly to converge in passing from the outer chamber, where it is comparatively at rest, to the point of entrance to the wheel chamber, where its velocity should be greatest. The advantage of such convergent channels for the transmission of water with a minimum of loss of effect, as compared with short passages such as are generally employed in other turbines, is well known.

Further, from the same principle of injection towards the centre there is an accordance between the velocities of all parts of the moving wheel and the proper velocities of the water in its passage between the points of entrance and discharge.

The water when it has its greatest velocity is admitted to the circumference of the wheel, which is the most rapidly moving part, and when it has as far as possible imparted its power to the wheel it leaves at the centre, which has the least motion.

The water enters from the guide passages with the velocity at which the outer circumference of the wheel is moving and without change of direction, so there is no loss from impact.

The steadiness and regularity of the motion of the vortex are remarkable, consequent upon the action of the centrifugal force of the water, which on any increase in the velocity of the revolving wheel, increases and so checks the supply entering from the guide passages, and on any diminution of the velocity of the wheel, diminishes and admits the water more freely, thus counteracting in degree the irregularities of speed arising from variations in the work to be performed. With other turbines and ordinary water wheels the variation in speed, when the amount of work or resistance to the motion varies, frequently is a source of great inconvenience.

Besides the adaptations referred to, which combine to give the vortex a great superiority in efficiency, it possesses another advantage, which is in many situations of the highest importance, namely, the simple and excellent mode of adjustment to varying supplies of water by means of the movable guide blades.

The guide blades, as already mentioned, are made movable upon gudgeons or centres near their points, motion being imparted to them simultaneously by a hand wheel, which can be placed in any position easily accessible. By a very slight motion of the guide blades the orifices may be contracted at pleasure, and are thus made to suit any quantity of water which it may be necessary or desirable to use ; and it will be seen that, to whatever extent these are open, the following important conditions of efficient application of the water are fulfilled :—

1st. The channels are of a gradually convergent form.

2nd. The water is uninterrupted in its course, and enters the wheel chamber from the narrowest part of the channels, and consequently attains its maximum velocity at the point of application.

3rd. The water is admitted equally to the whole circumference of the wheel.

Fig. 72 is taken from a photograph of a vortex wheel with the outer cover removed to illustrate the form of the vanes. Some of these do not extend to the central orifice; the object in so making them is that they may not too much fill up the contracted part of the passages and thus impede the flow of the water.

The wheels (Fig. 73) are constructed either of steel or of rolled brass (the latter for small sizes), and as the vanes can thus be made very much thinner than of cast metal their number can be increased, and perfect accuracy in the curvature secured. Hence the water enters the wheel with less interruption and passes through more exactly in the direction intended than is the case where the vanes are of greater thickness and fewer in number.

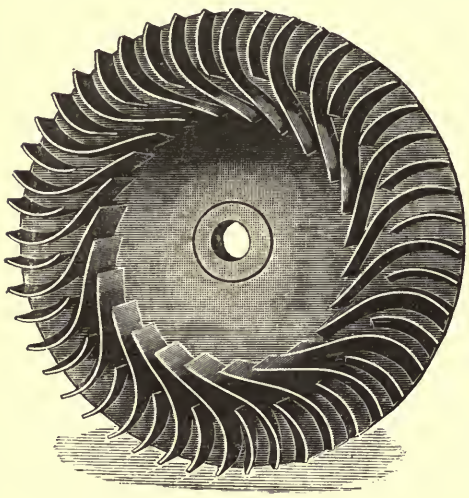


Fig. 72.—Vortex Turbine Wheel.

The vanes are fixed on each side of a steel or brass plate, which has a boss in the centre to secure it upon the shaft, and there are two outer discs or covers in which are left circular openings through which the water passes after it has done its work; thus one-half of the water is discharged on each side of the wheel.

A separate movable cover is placed over the wheel, so that access to it can be had without disturbing the exterior casing. A cover is also placed over the guide blade chamber in the larger wheels, by means of which access can be obtained to every part without moving the foundations or heavier portions of the case.

The double vortex, with movable guide blades, should be adopted on all medium and high falls in cases in which the amount of power employed varies considerably at different times; and the saving of water is important, so that it is necessary to use as small a quantity as possible to do the work required; or when the available supply of water is at times less than the full amount for which the turbine is designed. The consumption of water can then be economised to the utmost, as the passages can be regulated to admit only the exact quantity needed to do the work or to suit the available supply.

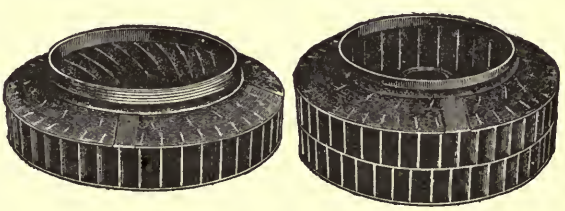


Fig. 73.—Vortex Turbine Wheels.

The governing of turbines is in many cases more difficult than the governing of steam engines, the two chief difficulties to be overcome being the weight of the moving parts and the fact that if the guides or sluice by which the governing is effected are worked too quickly very considerable hydraulic shock takes place, which may do damage to the pipe line. There are many governors in the market; most of these will give fair results to prevent racing, that is to say, they will not allow the turbine to run away so much as to do damage to ordinary machinery, but where great regularity of speed is required under varying loads,—for example, where a dynamo is supplying the electric current direct on to lamps which are switched in or out from time to time,—it is necessary to provide a sensitive governor which will not admit of perceptible fluctuations in speed.

Our experience of Mr. Murray's hydraulic governor enables us cordially to recommend its adoption for electric lighting plants such as are alluded to above, or

for any other purpose in which extreme steadiness is essential. With this governing apparatus—a full description of which will be found below, the water is, by the action of a high speed governor, turned to one end or the other of a hydraulic cylinder which works the guide blades. Such a governor will be seen in Fig. 74, Nos. 1 and 2 being sections at right angles to one another of a 4-way valve forming a part of the apparatus, and No. 3 an elevation showing the complete arrangement. In Nos. 1 and 2 the outer casing A is provided with a supply port B, an escape port C, and other ports D and E leading respectively to the top and bottom of a regulating cylinder which controls the supply to the motor. The action of the apparatus may be seen from No. 3, in which I is a centrifugal spring governor driven from the turbine and connected by the rod J with a lever K, to which the piston H is connected by a link. When the turbine speed increases the piston is raised and water passes by pipe D to the upper part of the regulating cylinder L, and acts on a piston M to reduce the supply. Should

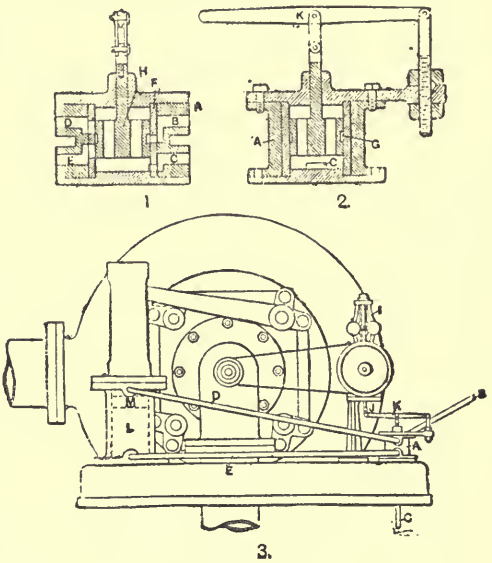


Fig. 74.—Murray's Governor.

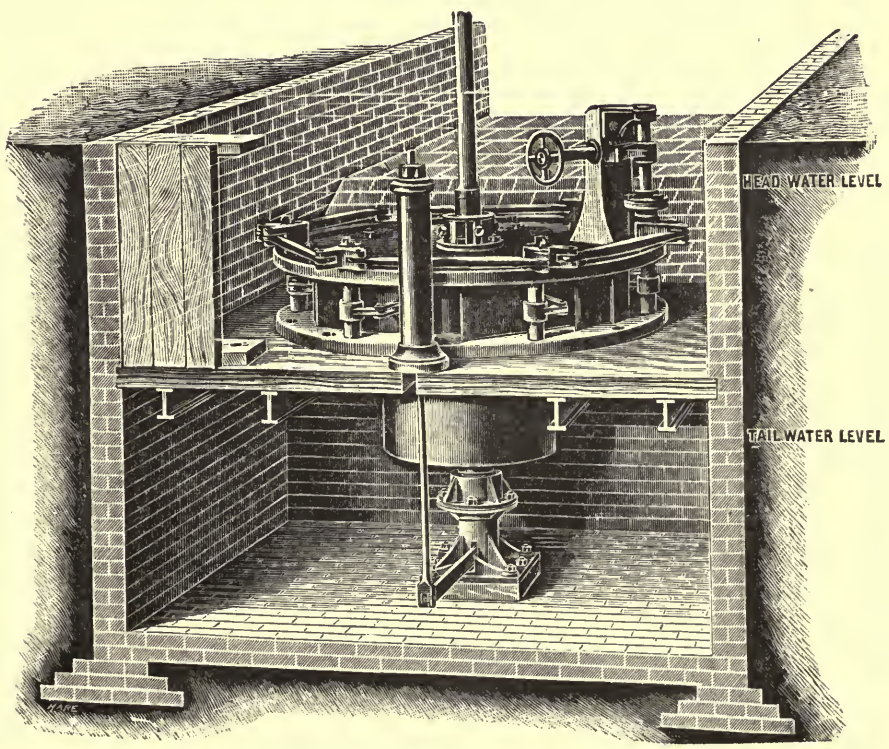


Fig. 75.—Method of Fixing Vortex Turbine.

the speed decrease the water passes by pipe E to the lower part of the cylinder L and

raises the piston, and permits more water to enter the turbine so as to maintain normal speed.

The essential difference between the single and double vortex turbines is that in the former the water is only discharged from one side of the wheel; the wheel of the single vortex being, in fact, half of that of the double, the centre plate of the latter forming the top cover of the former.

The guide blades direct the water on the wheel in precisely the same manner as they do in the double vortex, and may be either fixed or movable as is desired.

This single vortex turbine is very well suited to medium and low falls, where a considerable body of water is to be dealt with, and where it is desirable to have the turbine left dry when the head water is shut off.

As the water is only discharged below the wheel, and not above and below as in the double vortex, part of the fall may be utilised by a suction pipe.

When difficulty or expense is apprehended in sinking the foundations a considerable

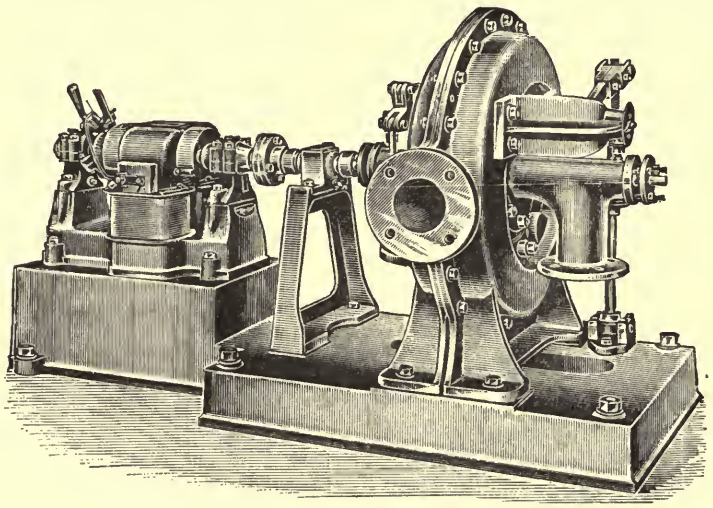


Fig. 76.—Vortex Turbine Direct Coupled to Dynamo.

saving may be expected, as the turbine can be placed on a platform above the tail water, and no work need be done below the water level beyond the sinking of a hole for the suction pipe to discharge into.

The single vortex is, of course, only to be used when the wheel is placed horizontally with a vertical spindle.

It will be seen from the illustrations that the heavy cast-iron casing that is required for the double vortex is dispensed with.

The usual method of fixing the single vortex is shown in Fig. 75. The wheel is placed on a floor made of timber or metal in a pit of masonry or brickwork. The timber breasting (which for convenience is shown broken away in the engraving) is of the same height as the sides of the watercourse. There is no way for the water to pass from the chamber above the floor to the tail race below except through the turbine.

The diameter of the single vortex is naturally larger than that of a double vortex for the same power; but so great are the advantages of dispensing with the outer casing and keeping the turbine above the tail water level that, where the quantity of water is large and fall low, it is adopted.

As the wheels are comparatively small in diameter, the rotational speed is pretty high. The speed of the wheel is some large fraction of the speed ot the water as it would flow out ot the wheel were it held stationary; and being closed all watertight,

it can very readily be coupled to a dynamo direct, and a great many of these vortex turbines are so used. Fig. 76 represents one of small size so adapted.

Below we give some particulars of interest regarding the single vortex wheel, from which the quantity of water required, the speed, and diameter of guide blade channel can be obtained.

TABLE IV.

H.P.		Fall in Feet.				
		8	9	10	11	12
10	Cubic feet per minute	883	784	706	642	588
	Revolutions per minute	91	106	118	128	137
	Approximate weight (cwts.)	53	47	43	41	39
	Diameter of guide blade channel (inches)	63	61	59	58	57
15	Cubic feet per minute	1325	1176	1059	962	883
	Revolutions per minute	80	89	99	107	118
	Approximate weight (cwts.)	63	59	55	53	49
	Diameter of guide blade channel (inches)	67	66	64	63	61
20	Cubic feet per minute	1765	1470	1412	1284	1176
	Revolutions per minute	71	83	89	96	106
	Approximate weight (cwts.)	73	65	63	61	57
	Diameter of guide blade channel (inches)	73	68	67	66	65

The particulars for double vortex wheels with horizontal shaft are—

TABLE V.

H.P.		Fall in Feet.						
		36	38	40	45	50	55	60
15	Diameter of inlet (inches)	10	9	8	8	7	7	7
	Cubic feet per minute	118	106	88	76	66	59	53
	Revolutions per minute	810	960	1055	1137	1389	1474	1553
20	Diameter of inlet (inches)	11	11	10	9	8	8	8
	Cubic feet per minute	157	141	118	101	88	78	71
	Revolutions per minute	768	810	936	1137	1216	1290	1359
25	Diameter of inlet (inches)	12	12	11	10	9	9	8
	Cubic feet per minute	196	176	147	126	110	98	88
	Revolutions per minute	730	769	886	1011	1216	1290	1359
30	Diameter of inlet (inches)	13	13	12	11	10	10	9
	Cubic feet per minute	235	212	176	151	132	118	106
	Revolutions per minute	644	680	842	958	1080	1290	1359
40	Diameter of inlet (inches)	15	15	13	13	12	11	11
	Cubic feet per minute	314	282	235	202	176	157	141
	Revolutions per minute	595	629	744	910	1024	1086	1144
50	Diameter of inlet (inches)	17	16	15	14	13	12	12
	Cubic feet per minute	392	353	294	252	221	196	176
	Revolutions per minute	592	629	744	803	973	1932	1087

TABLE V.—*continued.*

H.P.		Fall in Feet.						
		36	38	40	45	50	55	60
60	Diameter of inlet (inches)	19	18	16	15	14	13	13
	Cubic feet per minute .	470	424	353	303	264	235	212
	Revolutions per minute .	522	549	683	803	859	1032	1087
70	Diameter of inlet (inches)	20	19	18	16	15	15	14
	Cubic feet per minute .	549	494	412	353	309	275	247
	Revolutions per minute .	476	502	602	737	788	910	960
80	Diameter of inlet (inches)	22	21	19	17	16	15	15
	Cubic feet per minute .	628	564	470	404	353	314	282
	Revolutions per minute .	476	502	549	649	695	837	882
100	Diameter of inlet (inches)	24	23	21	19	18	17	16
	Cubic feet per minute .	784	706	588	504	441	392	353
	Revolutions per minute .	398	462	505	580	635	673	706

The following construction is used in setting out the blades in an inward radial flow turbine.

The guide and wheel blades for inward flow turbines are partly involutes of a circle, to lessen the contraction of the stream as it flows through the passages. Fig. 76*a*, A is the external circumference of the guide apparatus, and B the internal circumference; draw the perpendicular CD. Make the angle CBE 15°, and draw the line CE at right angles to BE, then describe a circle with C as centre, through the point E. Let BF be the inner pitch of buckets, and suppose a thread wound round the circle E, and carrying a pencil at B; as this is unwound, the arc BGH is traced by the point B, the point H being a little to the left of the line JFG, and the width of the passage being, therefore, uniform from G to H. The remainder of the bucket HA is a portion of a circle, the tangents AD and AK containing the angle 87.5°. The buckets in the revolving wheel are constructed similarly, except that the angle L is made 12° instead of 15°, and the angle M, instead of 87.5°, should be made an angle of 90°.

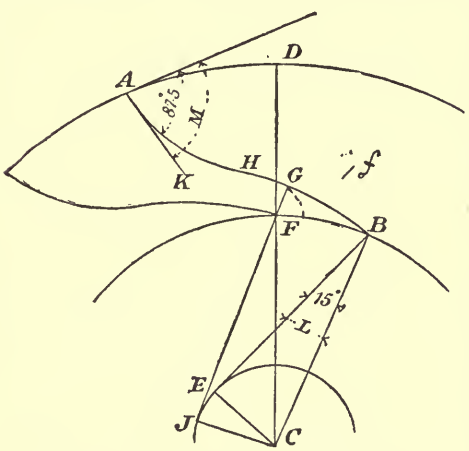


Fig. 76*a*.—Diagram to find Curvature of Turbine Blades.

In some constructions the number of guide blades are made equal to or nearly equal to the number of wheel blades. In the vortex we have only four or six guide blades.

The leading dimensions of a vortex wheel can be found as follows:—

Let Q = cubic feet of water per second, as before.

h = the effective head.

V = velocity in feet per second due to h.

V_1 = ,, ,, ,, in guide blades.

V_2 = ,, ,, ,, in inner orifies.

Let A_1 = cross section guide blade orifice at discharge.
A_2 = ,, ,, wheel passages.
r_1 = radius of circumference on which water enters wheel.
r_2 = ,, ,, ,, ,, leaves wheel.

Now, in setting out upon a design we have given the value of Q and h. From this r_1, r_2 can be calculated if we fix upon a limiting number of revolutions per minute.

There are other mechanical calculations, such as the breadth and depth of the blades and their number and their combined sectional opening.

Then let C_2 = the velocity of inner radius of wheel, and
C_1 = ,, ,, outer radius of wheel.

Then it is found that the area A of the guide passages should be 1.15 times the area A_2 of the wheel passages, and that r_1 should be 1.17 times r_2. And if there is a draught tube, its area A_3 should be 1.25 times the area of A_2.

We can now proceed with the design in figures, starting with the given data for which a turbine is required—

$Q = 13$ cubic feet per second.
$h = 36$ feet effective.
$V_1 = 0.57 \times 8\sqrt{h} = 4.56\sqrt{36} = 27.36$ feet per second.

Now, from construction the velocity C_2 at the inner radius equals $V_1 \times 0.966$ $\therefore$ $C_2 = 27.36 \times .996 = 26.44$ feet per second.

Then, we can settle the value of r_1 and r_2 if we know what speed in revolutions per second is required. We will assume the speed required is 420 per minute, or 7 per second; and if we now divide V_1 by speed per second we get the circumference at r_2, and from the circumference the value of r_2 from a table; hence $\frac{26.44}{7} = 3.77$ feet, or 45 inches circumference. From a table of circumferences and diameters we find the diameter corresponding to this circumference = $14\frac{1}{2}$ inches. Again, to find r_1, the outer radius, a rule in practice arrived at by experience is $r_1 = r_2 \times 1.17$, and as $r_2 = 7.25$, r_1 will equal $7.25 \times 1.17 = 8.6$ inches, or outer diameter = 17.2 inches. That rule holds good for inward flow wheels with multiple guide blades,—nearly as many as wheel blades,—but for the vortex the rule is nearer $r_1 \times 1.5 = 11$ inches nearly, or 22 inches diameter. Thus we have arrived at the speed in revolutions, and the diameters of the wheel. We find the combined area of the wheel passages at the inner radius. This is equal to $A_2 = \frac{Q}{V_2}$; hence $= \frac{13}{26.44} = 0.5$ square feet, or 72 square inches clear opening of combined wheel passages.

The circumference at r_2 is = 45 inches. The number of blades is determined by the circumference divided by 1.5 in inches; hence $\frac{45}{1.5} = 30$ blades. And if each blade is $\frac{1}{8}$ inch thick we get 3.75 inches as space occupied by these blades by their thickness on this circumference of r_2 and the contraction due to the angle, as shown by the figure. If we divide off FB, using F as a centre with radius FG, then the radius Ff cuts off on FB a part equal to the opening, and FB is to Ff as the whole circumference is to the actual opening; so that if the distance Ff is 0.25 of FB, then the actual opening is as .25 to 1, or $45 \times .25 = 11.25$ inches. And deducting 3.75 for thickness of blades, we get $11.25 - 3.75 = 7.5$ inches as A_2 on one side; dividing the total area .5 square feet by 7.5 we get $\frac{72}{7.5} = 9.6$ inches as the axial breadth of the blades at the inner radius.

The guide blades, if numerous, must be calculated in the same way.

Again, the ratio between the radius r_1 and r_2 we took as 1.5 to 1, so that the breadth of the wheel axially may be reduced outwards in that ratio. Hence, since the blades

are 9.6 inches broad at r_2, they will be $\frac{9.6}{1.5}$ at $r_1=6.4$ inches. This reduction at the outer periphery is shown in Figs. 72 and 73 very well.

The rest of the design is mechanical, such as the casing, the bearings, shaft, and governor; these come under other headings. The inward flow turbine is sometimes made as shown in Fig. 77, an illustration of a common form, in which the wheel passages are spiral and the discharge axial.

Fig. 78 shows how this turbine is conveniently fixed in the pentrough.

Another type of mixed flow turbine is well shown by Messrs. Gunther's illustration of a large pair recently made.

We must consider the so-called mixed flow turbine, to which a large class belong. Usually they are inward and axial flow, and can be easily placed in a timber or brick wheel pit with the draught tube projecting through the floor.

In some of them the guide blades are made movable for regulating purposes, while some have a revolving cage whereby the orifice can be reduced in area. The wheel, usually of gun-metal or bronze, is shown in Fig. 77. The blades are curved and

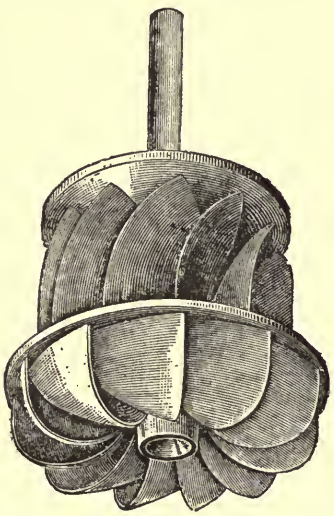


FIG. 77.—Mixed Flow Turbine.

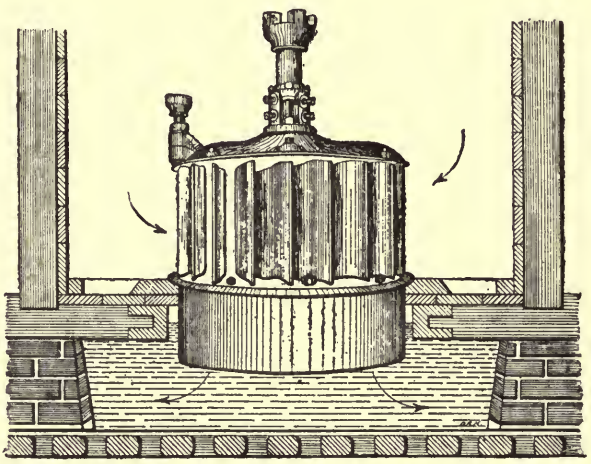


FIG. 78.—Method of Fixing Turbine.

spiral, leading down to outward flow; curved also for downward flow blade. These types of turbines are all difficult to regulate in speed, but where close regulation is not of importance they are cheap to instal, and give a good efficiency when full loaded.

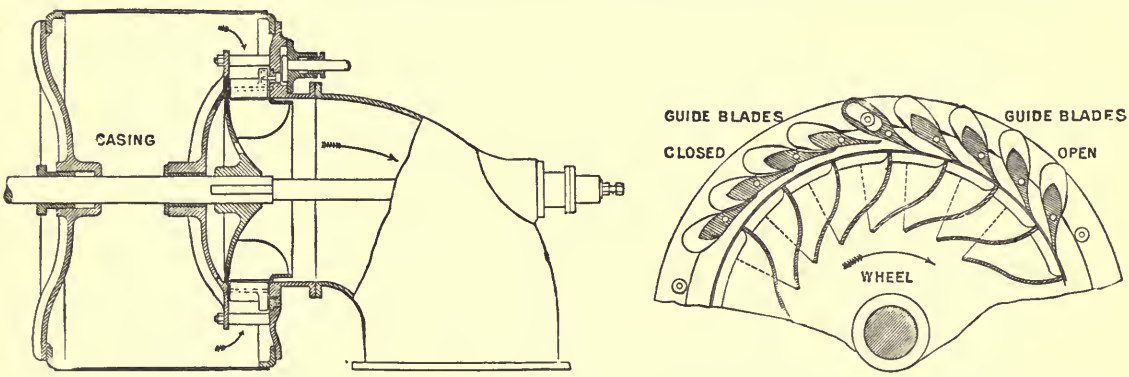
The mixed flow turbine constructed by Messrs. Gunther has an axial and outward radial flow set of blades, with a series of guide blades acting much as the large blades in the vortex wheel. As it is a good type of wheel for a horizontal shaft and considerable speeds at high powers, we may consider a good example of it recently made for a large mill abroad.

The plant consists of two of the firm's latest type of mixed flow turbines, in which the water enters on the outer circumference of the wheel, and is discharged partly on the inner circumference and partly in an axial direction. The head of water available is 35 feet, and each turbine has been designed to produce 150 horse-power with this head at a speed of 180 revolutions per minute. Plate I. shows the complete plant mounted on its bed-plate, consisting of two cast-iron beams, which, when everything is bolted up, form what is practically a continuous bed-plate. It will be seen that the two turbines are placed back to back with their shafts in alignment, and that there

is a rope pulley mounted between bearings on a small shaft in a line with the other two shafts. It may be connected to either or both turbines by means of two claw clutches worked by levers, as may be seen in the illustration.

A question which readily suggests itself when looking at the combined plant is, Why were two turbines used? In these days a 300 horse-power turbine is comparatively small, and size would have been no obstacle; moreover, the expense of a double set of piping would have been saved. In this particular instance, however, it was deemed preferable to have two instead of one, and for two principle reasons. The first of these had reference to the weight of any single portion of the machinery. Owing to the locality for which the plant was destined it was desirable to keep down weights as much as possible. Then, too, one turbine can be used alone when the full output of 300 horse-power is not required.

The rope pulley is 5 feet 3 inches in diameter, and it is grooved for eighteen ropes, each $1\frac{1}{4}$ inch in diameter. The turbine wheels are 36 inches in diameter, and the general construction may be seen in Fig. 79, which shows a section through the turbine case, and a section of the guide and wheel vanes, showing the position of the former when closed and when fully open. The casings are of mild steel plates with cast-iron flanges riveted in, and each turbine is provided with a manhole



Sectional Elevation. Section of Guide Blades and Wheel Vanes.

FIG. 79.—Large Mixed Flow Turbine.

for examination of the interior. The water inlets are 42 inches in diameter, and the suction bends 36 inches in diameter. The wheel of either turbine can be taken out by removing the suction bend and the cast-iron end to which it is bolted, which does not necessitate touching anything on the pulley side. The end pressure of the shaft is said to be very slight, and to take it up a lignum vitæ footstep is provided. This is placed in a small chamber, which may be seen bolted to the outside of the suction bend. It is adjustable for wearing by means of the set screw at the end of the chamber, and it is lubricated by the pressure water taken from the turbine case by means of a small pipe, which may be seen in the engraving. This lubrication is so arranged that the pressure water passes up the centre of the footstep and through a groove across its diameter. It is said that this construction ensures that the whole rubbing surface is efficiently lubricated. As the pressure water might contain grit or sand, measures are taken to prevent this getting into the bearing.

Regulation for speed or power is, as may be gathered from Fig. 79, brought about by movement of the guide blades. The guide blades are pivoted, and are opened or closed simultaneously by the rotation of a ring carrying a number of equally placed pins, one of each fits into a slot in each guide plate. Quite a small motion, or travel of the ring, suffices to fully open or fully close the blades. By properly proportioning the guide blades, and placing the pins on which they work, matters are so arranged that

the blades are practically in working balance against the pressure of the water, and that consequently the gates are easily and quickly opened. As a result close governing is obtained. The ring which opens or closes the guide blades is worked by two

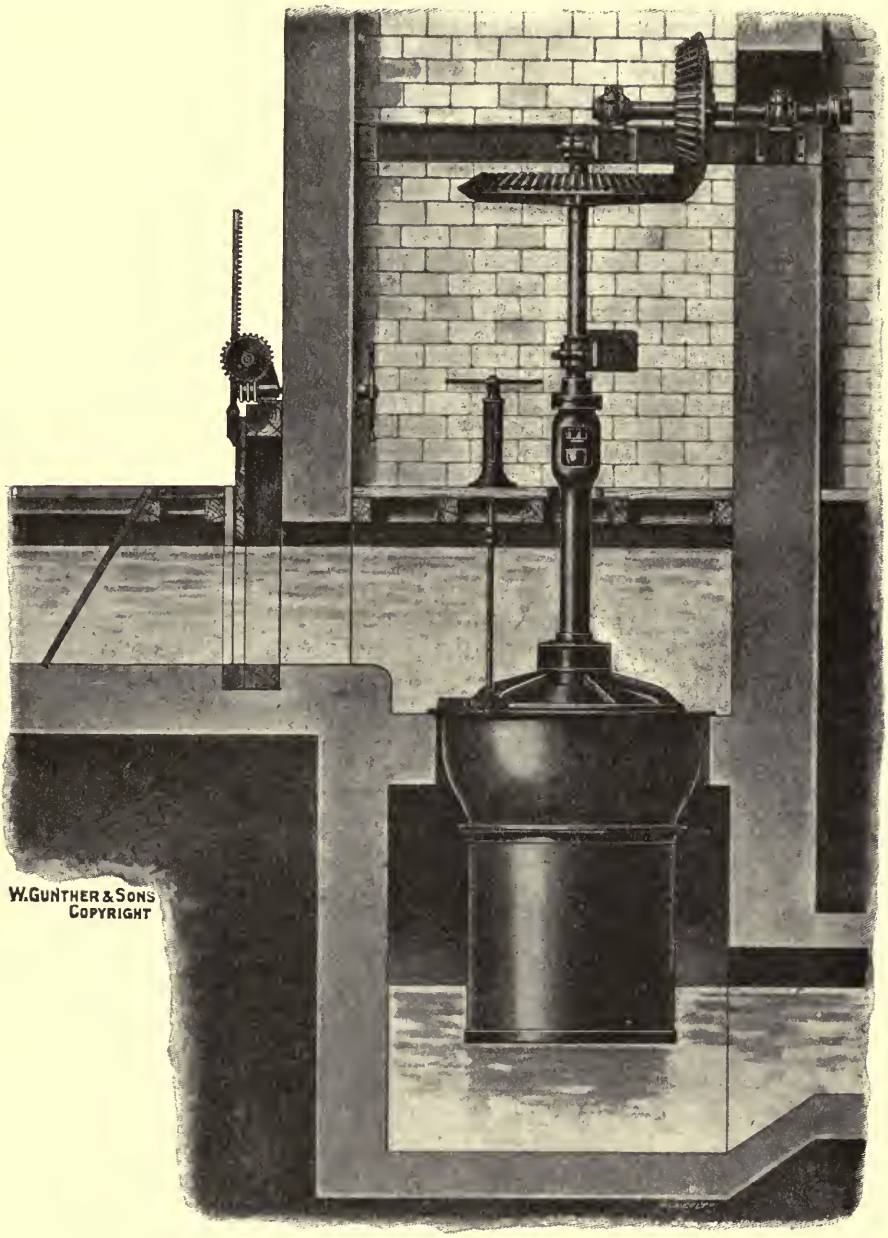


FIG. 80.—Method of Fixing Large Jonval Turbines for Low Falls.

cranks placed 180° apart and coupled by levers, the desired motion being imparted by the screw revolution of a screwed spindle.

The spindle actuating the regulating levers is provided with a coupling at its upper extremity. This is taken to an upper floor over the turbine house, and there connected to the regulating pillar and governors.

PRESSURE TURBINES

These are inward or outward flow pressure turbines usually, but are also made for axial flow.

This is the type ot wheel used at Niagara Falls in the large power scheme, working without a draught tube, and they are outward flow. We shall refer to them again.

But before the Niagara scheme there was a paper mill in Niagara with a wheel-pit 167 feet deep and 28 × 43 feet in section, with a vertical supply pipe 13½ feet in diameter, working three inverted Jonval turbines of 1100 horse-power each. By inverted is meant that the guide blades were below and the wheel above, so that the upward water pressure supported the weight of the wheel and vertical shaft, the shaft being 10 inches diameter.

Niagara Falls are on that immense chain of inland lakes on the boundary between Canada and the United States; these lakes are natural reservoirs for the rainfall over 240,000 square miles. The overflow discharges at a constant rate nearly, through the river St. Lawrence into the sea. Between Lake Erie and Lake Ontario the water flows through the river Niagara with a fall of 326 feet in 36 miles, the quantity of flow is about 300,000 cubic feet per second; such a vast quantity at such a high fall would, if it could be utilised, give up about 7,000,000 horse-power night and day.

In all these large powers some modification of the pressure wheel is used with a vertical shaft.

The arrangement for setting large Jonval turbines in masonry for low falls is shown in Fig. 80. The water enters from the left and first passes through a screen, to intercept floating rubbish; it then passes the sluice valve into the wheel pit and falls through the wheel into the tail race. And Fig. 81 shows an axial turbine of this class with a horizontal shaft made in a convenient form for coupling to a draught tube. Again, for an inward flow turbine, let—

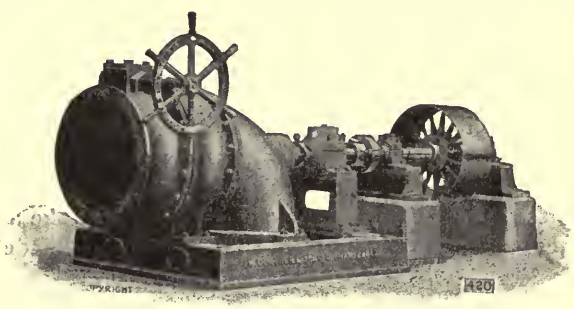


FIG. 81.—Jonval Turbine with Horizontal Shaft.

Q = cubic feet of water available per second.
h = effective head of water.
V = velocity of water due to fall in feet per second.
V_2 = ,, ,, across clearance between guide blades and wheel in feet per second.
C = velocity of wheel blades at entrance of water at r_1.
C_2 = ,, ,, ,, outlet r_2.
A_1 = sectional area of jet at discharge of guide blades.
A_2 = ,, ,, wheel passages.
A_3 = ,, ,, tube or exhaust.
R = revolutions of shaft.
r_1 = inner radius of wheel.
r_2 = outer radius of wheel.
N = number of wheel blades.
N_g = ,, guide blades.

Empirical rules give the following data:—

$$\frac{r_1}{r_2} = \frac{1}{1.17}.$$

$$\frac{A_2}{A_3}=0.25. \quad V_2=0.57\times 8\times \sqrt{h}.$$

$$C_2=0.96\times V_2. \quad A_2=\frac{Q}{V_2}.$$

Assume we wish to calculate a wheel for 100 feet head effectual. Quantity of water $Q=50$ cubic feet per second; speed of wheel, 10 per second. We shall proceed to find the dimensions of the wheel. Thus from $V_2=0.57\times 8\sqrt{h}=0.57\times 8\sqrt{100}=4.56\times 10=45.6$ feet per second. And C_2 the velocity of the wheel at r its outer periphery in an inward flow turbine $=C_2=0.96\times V_2=0.96\times 45.6=43.7$ feet per second.

From this and the speed 10 revolution per second we get r_1; for $\frac{C_2}{R}$ will give us the circumference in feet, from which r_1 can easily be found. Thus $\frac{43\cdot 7}{10}=4.37$ feet circumference $=4.37\times 12=52.44$ inches, equals a diameter of 16 inches; $\therefore r_1=\frac{16}{2}=8$ inches, and $r_2=\frac{8}{1.17}=6.8$ inches, and 8 inches for outer r_1 in an inward flow wheel.

The combined area of the guide blade passages $=\frac{Q}{V_2}=\frac{50}{43\cdot 7}=1.15$ square feet $=165$ square inches $=A_1$.

From this and the number of blades and their thickness the axial depth of the wheel can be calculated, for we know its circumference at r_1. But besides the thickness of the blades and their number we want to know their contraction of passages by their obliquity to the radial line; this can be calculated if we know the angles. The angles will be, however, better drawn to instructions given in Fig. 76*a* (p. 60) already, and the actual opening F G measured; before drawing it we must know the number of blades. These should be numerous and thin. If we take a safe estimate from practice of making them about 1.5 inches from the centre of one blade to the next on the circumference, this circumference is 52 inches; hence $\frac{52}{1.5}=35$ blades. Each blade might be $\frac{1}{8}$-inch thick; hence the lot will take up $\frac{1}{8}\times 35$ about $7\frac{1}{2}$ inches. This amount must be deducted from the circumference.

We must find the openings on the diagram, Fig. 76*a*, F G, and suppose we find them each 0.25 of what they would be if the blades were truly radial, that is to say, the openings would be only $\frac{1}{4}$ of what they would be with radial blades, $\frac{1}{4}$ of $52=13$ inches, and this $-4\frac{1}{2}$, the thickness of blades $=8\frac{1}{2}$ inches as the one side of the area of the outflow. $A=165$ square inches; hence $\frac{165}{8.5}=20$ inches, nearly the axial depth of the wheel. We have now found—

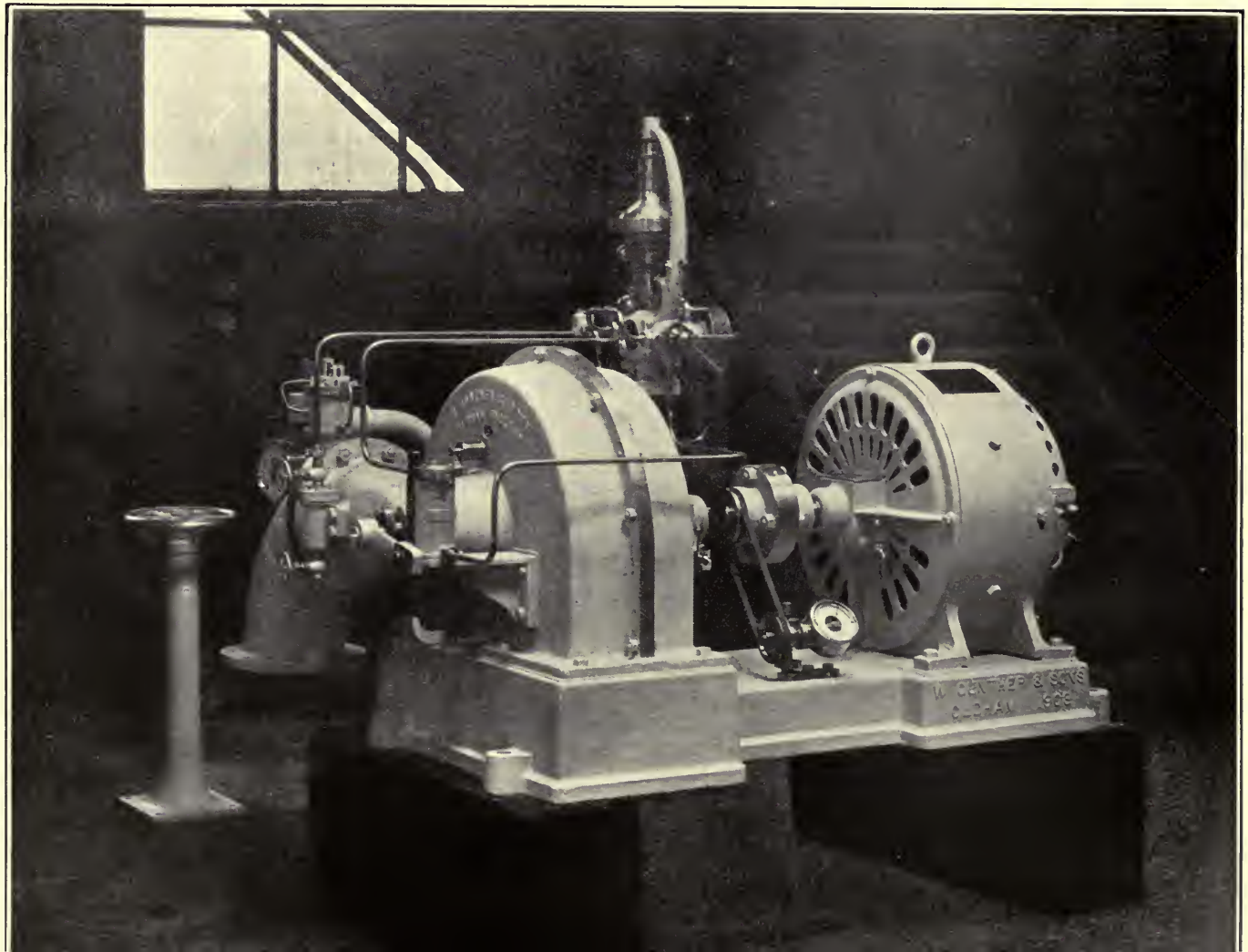
Diameter of wheel outside 16 inches.
,, ,, inside 13.6 inches.

Area of guide blade passages $=165$ square inches, and the number and thickness of blades from which we find the length axially of the wheel.

If a draught tube is used its area should $=A_1\times 1.25$.

On this basis there would be some pressure, but very little where the water jumps from guide blades to wheel blades, for in an inward flow turbine it is difficult to get the r_2 end of the opening larger than the guide blade openings; they must be smaller.

In the axial flow Jonval turbine the radius r is the mean between the outer and inner circle filled by the wheel passages. Here the passages A_1, A_2, A_3 are easily found, thus $A_1=\frac{A_2}{1.15}$, and $A_3=\frac{A_2}{0.25}$; so that if $A_1=1.15$ feet, A_2 would $=1.15\times 1.15=1.8$ square feet. And $A_3=1.8\times 4=7.2$ square feet opening.



TURBINE VANES

In the foregoing calculations values and constants have been taken to show the working of elementary problems in turbines. The subject, however, can be treated with mathematical precision by geometrical, graphic, and algebraic figures and formulæ to be found in classic works. While the conclusions to be arrived at by these means are sure guides as to the practical designing and construction, they cannot be realised entirely.

In dealing with guide bladed turbines, in which the energy is delivered mostly by momentum, we speak of velocities as "absolute" and "relative." To explain the meaning of the terms we may refer to the Pelton jet and wheel (Fig. 50). The nozzle is fixed, and the jet issues at a velocity due to pressure. This velocity is absolute; it is the velocity with which a particle in the jet moves away from the nozzle to the vane; "relative" velocity means the velocity of something else compared with this absolute velocity. The vane, for instance, might have no resistance to offer to the jet, in which case it would be carried forward with the jet at the same velocity, and the relative velocity ot the vane would be the same as the absolute velocity. Again, if the vane were fixed there would be no relative velocity between it and the jet. In the first case the jet would not split and flow back out of the vanes; the vanes would be carried along like a shield on the advancing end of the jet; being unresisted, no work could be got from it. In the second case the water would split, and, if we neglected friction, would return with same velocity, exerting a pressure only on the vane. Now, between these extreme cases we have relative velocities at which we obtain motion of the vane against resistance, and there is a certain velocity of the vane compared with that of the jet from which we obtain the greatest amount of energy, and that is a relative velocity midway between the two extremes which we have considered,—that is, when the velocity of the vane is half the velocity of the jet. When the vane had no resistance against the jet its velocity was equal to that of the jet, and therefore no pressure could be obtained upon it. When the vane was at rest full pressure was obtained, but no velocity. If now the vane is allowed to move at half the jet velocity the maximum work done against the resistance which controls the relative velocity for the pressure × the vane velocity = the power obtained. As V the relative velocity of the jet increases, P the pressure upon it decreases; and as P the pressure on the vane increases (by restraining the vane to a lower V), V decreases. Take a rectangular parallelogram (Fig. 82), divide the top line to represent parts of pressure on the vane, and the bottom line to represent parts of velocity, numbered from right to left for pressure, and from left to right for velocity; a perpendicular line anywhere between the two ends will cut the area into two parts corresponding to V and P respectively in magnitude, so that if we increase the one area we decrease the other in the same amount. Suppose we divide it at 5 on bottom line, then P = 15 and V = 5; hence P × V = 75. Divide it at half, that is, make P = V, then 10 × 10 = 100. Any other division than half will give a less result; hence in a Pelton wheel the relative velocity is half the absolute velocity for best effect.

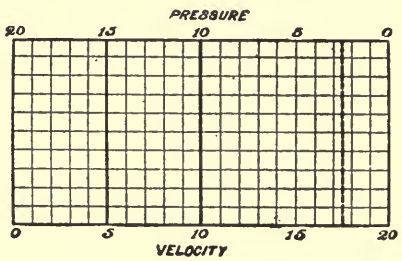


FIG. 82.—Diagram showing Relation between Pressure and Velocity.

This is due to the jet striking squarely on the vane in the direct line ot motion, and the curves deflecting the split jet to clear the incoming jet and effect the pressure. The jets leaving the vane have no relative velocity, for the vane is receding from them in the opposite direction to that which they flow in.

But the case is different if the jet strikes the vane at an angle and leaves it at

an angle. The figure representing the effects is then a rhomboid, with the sides drawn to scale showing the value of velocities.

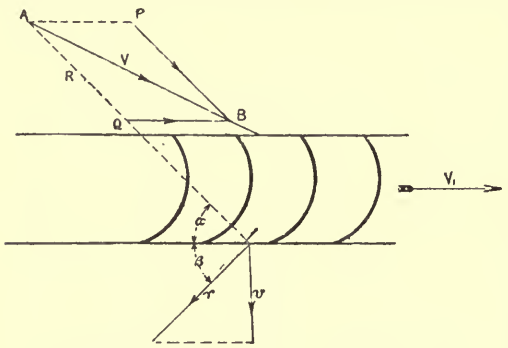


FIG. 83.—Diagram of Forces acting on Turbine Blades.

Suppose in Fig. 83 a part of a turbine wheel. The jet enters along line V, its length representing its velocity.

We must resort to the resolution of forces by the parallelogram of forces.

Suppose for illustration we take a model to illustrate the motions of a body like an inclined plane under a force acting at an angle. It is not sufficient in practice merely to represent the direction and magnitude of forces by lines and angles; it is better to see the effects first, and take the line and angle representation afterwards. Place a wedge with an angle of 60°, say, at A on a plane movable on rollers (Fig. 84), and have a guided sliding rod with a roller on the end, so that it can move easily up or down, but not sideways. Now, if we press with a force on this rod downwards there will be a certain amount of the force transmitted downwards, but owing to the incline some force will act from left to right, and the plane will run forward in the direction of the arrow, while the rod will move downward. And if the height of the wedge P from A to B is double the length CB, as it is, then the plane F would move twice as fast as the rod W.

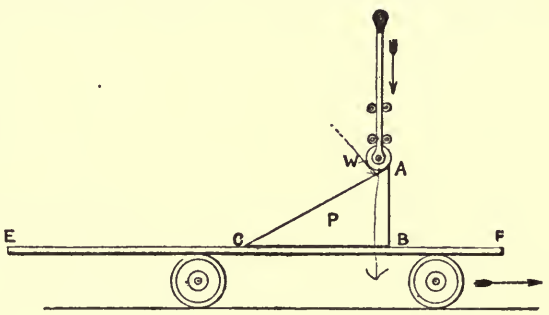


FIG. 84.—Experiment with Forces acting on Inclined Plane.

Now, if rod W is replaced by a water jet, CA may be considered the vane of a turbine; we would get the same effect, only the water would rebound from the plane.

Referring to the Fig. 83, the water travels along line V; then, if we draw a line QB equal in length to a scale representing the velocity V_1 of the vanes, then the relative velocity of the jet to the turbine will be =QB. Drawing this from the end of V parallel with the turbine motion, and completing the parallelogram APBQ, PB will give the relative velocity of the jet. The absolute velocities when entering and leaving the vanes are the important points, not the velocity of flow through the vanes, as the energy given up depends upon the square of these velocities, and the efficiency would be $\frac{V^2 - v^2}{V^2}$.

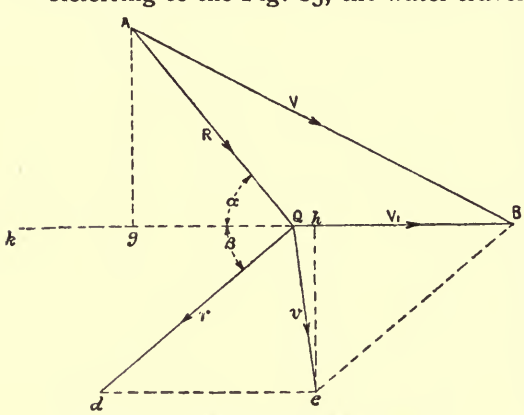


FIG. 85.—Diagram of Forces acting on Turbine Blades.

The velocity v must be reduced to the smallest value, and this depends on the velocity V_1 of the vanes and on the angle of entrance of the vanes and the angles of exit; this also can be shown by the parallelogram of forces. Fig. 85, AB represents, again, the direction and velocity of motion of the jet; QB represents V_1, the velocity of the

turbine blade in direction and magnitude; then join AQ by a line, this represents the velocity of the jet relatively to the turbine = R. Constructing the parallelogram QB*ed*, with angle α and β equal, we get Q*e*, the magnitude and direction of *v*, the issuing velocity of the jet at the exit. From this figure it is easy to prove that the smaller the angles α and β the smaller *v* would be, and that if the jet is completely reversed in direction $v = \frac{1}{2}V$. At best effect, and whatever these angles may be, this value is never less than $\frac{1}{2}V$, but greater the larger α is; but even with angles α and $\beta = 45°$ the lost effect is only 17 per cent., and in practice this angle varies between 15° and 22½°, so that the actual effect of a variation in the angles has not great influence.

To draw the vanes of a parallel flow guide, draw two lines parallel to represent the entering face and the exit face of the guides to any given scale (Fig. 86). Take FG = the pitch, and make the angle DFG = to α; from G draw a line GDA perpendicular to FD.

With centre A, radius AD, describe the curve DB; proceed with the others in the same way.

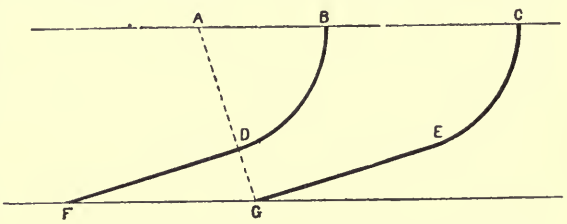


FIG. 86.—Diagram to find Curvature of Guide Blades.

The water is directed in the proper direction as it enters and gains velocity in the narrowing passage of the guide blades.

For the blades of inward flow turbines we have already given instructions on p. 61. For outward and inward flow wheels and all wheels a different construction is required for impulse and reaction wheels, as already described.

Fig. 87 represents a method for setting out the wheel blade curves. Draw two lines representing the entering and exit forces of the wheel, and divide off the pitch of the blades FG. And as before for the guides, raise a line from G perpendicular to FE, having made the angle of exit GFE = B. At an angle of 45°, from point E draw line EC cutting AD at C; make the triangle CBE equilateral; B will then be the centre of the curve from C to E.

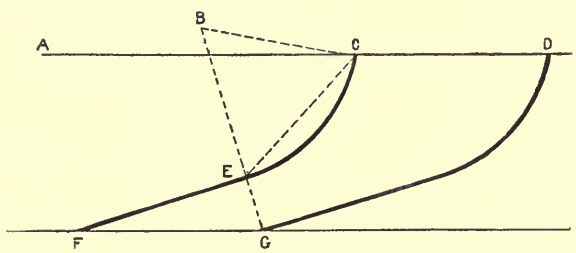


FIG. 87.—Diagram to find Curvature of Wheel Blades.

The designing of turbines, like the designing of all prime movers, is not altogether controlled by theoretical considerations. These are admirable guides, but practice and experience alone can produce good designs from these theoretical considerations.

Turbine makers and designers keep their experience among themselves, and have their own methods of working out details. The designer should keep in view the Pelton wheel as an elementary impulse turbine, also the Hero turbine, as the elementary pressure or reaction turbine as shown in Fig. 39, and also as shown in Fig. 40 drawn as an impulse turbine.

The form of blades shown in Fig. 34 of a Girard wheel is more of a curve than those seemingly dictated by theory, yet they are quite efficient.

We may now turn to the question of governing water turbines. This is no ordinary problem. The chief difficulties arise from the inertia of the water, its weight and momentum. To check an increase of speed due to fall of load, the water flow must be checked. If this is done suddenly great pressure results, and something may burst to prevent such an accident. Air vessels may be used on the mains to give some elasticity. Again, if the water requires acceleration to increase the speed when it falls

due to increase of load, time is required to increase its flow. Early governors were mostly designed of great power and magnitude, in order to operate valves or sluices on the main supply; these, being large and subject to great pressure, were very sluggish and useless for any work requiring steady driving.

In many cases, however, the speed was wonderfully steady, despite the poor governor; this was, however, due to the fact that the working load, which could be varied to any extent, was a small fraction of the total load. The mill gearing is usually machinery of a constant load, being in most cases about 75 per cent. of the total, so that if 100 horse-power turbine were in use its load could not have a maximum variation of more than 25 horse-power out of 100, and this could only occur in the most unlikely event of all the power on the variable load happening to go off at once, probably the variation in load on a turbine working a mill, through belts, pulleys, spur wheels, and shafting, never exceed 10 per cent. of the total. That being so, the governor in most cases was more ornamental than useful, and really acted more as a safeguard against dangerous racing in the event of a main belt or shaft failing.

All that, however, has been changed by electrical transmission of power, in which the variable load is 75 per cent. of the whole, and the constant load correspondingly small, and it is quite possible to throw off all the variable load at once by the opening of a switch or cut-out. Hence the *governor* must be made to *govern* the turbine speed promptly, and within a small variation. The known governors are not many, but present-day forms of them are fairly satisfactory, and close enough governing can be obtained by the best of them. Some turbines are easier governed than others. We have referred to the Pelton wheel governor and the vortex governor already in connection with these wheels.

TURBINE GOVERNORS

The question of turbine governing when constant speed under varying loads is one of much importance, and presents some difficulties when the load is liable to sudden and large fluctuations; especially is this the case when driving electrical generators for constant electric pressures. The same difficulty arose with steam and gas engines in early days, for until electrical engineering became a commercial success the governing of engines was not understood. The marine engine, the locomotive engine, the factory engine, traction engine, and others got along very well without any particularly good governing. In most cases of power users the power required was pretty constant, for the gear used to transmit it generally took from 40 to 70 per cent. of the maximum all the time. The marine engine cannot run above its maximum speed unless the propeller is lost or the propeller shaft broken, or in storms, when the propeller may be lifted out of the water. The locomotive is governed by the driver of the engine.

Steam, gas, and water powers are governed by centrifugal pendulum governors as a rule. This type of governor, originated by James Watt, is capable of considerable variation in design.

For water turbines, wherein the valves to be moved, or sluices to be shut, or blades altered in angle, a hydraulic relay is necessary. The old attempts to govern by powerful governors are all failures; mechanical relays also are failures. A light sensitive governor controlling a powerful cylinder and piston which operates the valves is the only solution of the problem at all satisfactory.

The centrifugal governors we shall treat later on fully under "Engine Mechanism." Meanwhile, the points of interest are the hydraulic principles used in the governors to operate the sluices or gates of the turbine.

In some old style turbines a large pendulum governor was used to throw in and out two clutches to gear into the sluices from the turbine shaft so as to open or shut them. But such governing always occupies too much time, and hence oscillations in speed

occur at every change of load. If the load is increased the speed decreases, and the governor engages the clutch which opens the sluices; but it takes time to open the sluice, so that the speed does not at once cease to fall, and the opening goes on until the speed has increased somewhat; it opens too much, so that the speed rises enough to throw in the other clutch to close the sluices; this closing is again overdone, and so the see-saw in speed goes on, gradually settling. A heavy fly-wheel modifies these oscillations.

The relay hydraulic governor shown in Fig. 88 was one of the first successful ones. A piston valve A with a free passage through the centre and an annular passage acts as a valve, admitting water to either one or other of two piston heads of different areas. On the smaller piston below there is a constant water pressure, tending to raise the piston if the water above the large piston is open to discharge through A and the overflow; hence if the governor raises the valve from the position shown the water above the large piston will escape, and the pressure on the small piston will raise it until the valve closes in the higher position.

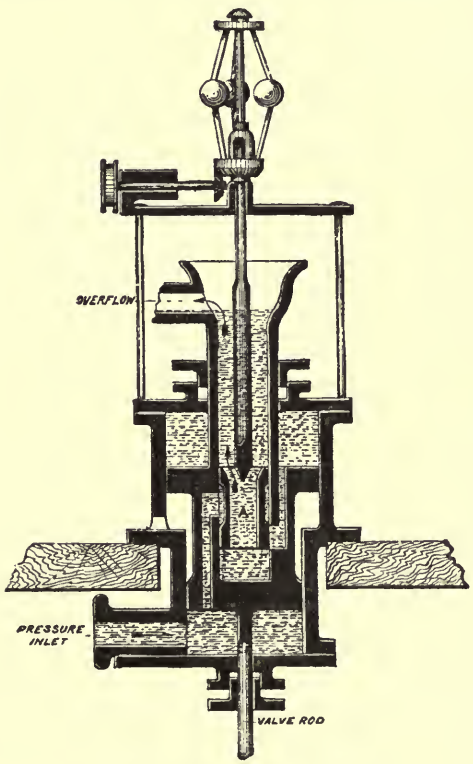


Fig. 88.—Relay Hydraulic Governor.

Again, suppose the valve dropped from the position shown, the pressure would be admitted to the larger piston head, so that the piston would drop as far as the valve had dropped, and would be locked in the new position.

The movement of the piston follows, and tends to close the port at whatever position the governor shifts the valve; hence the movements of the small light valve are followed by the powerful pistons as if the pistons were directly connected to the governor. The action is prompt and positive. The figure shows this governor in its elementary form, and as so constituted, is all that is required for sluices requiring small amounts of movement; for long movements the governor is connected to the valve rod by a lever, the fulcrum of which rises and falls with the piston.

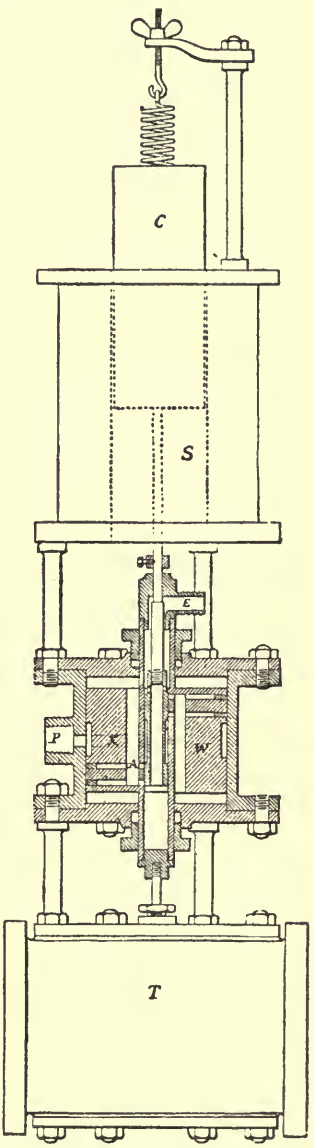


Fig. 89.—Willans' Governor.

In Murray's governor (p. 57) the valve and piston are separate, so that a long stroke can be made with a very short movement of the governor valve.

Willans' governor can be actuated by an electric solenoid operated from a dynamo when the turbine is used for electrical generating. The arrangement is shown in Fig. 89, in which the governor valve is inside the piston, which acts in the same way as described for Fig. 88.

Many attempts at solenoid governing have been made, and Willans contains the elements of success first achieved. Willans, instead of actuating the throttle valve or expansion valve directly by the electromagnet or solenoid, employs the latter to actuate a small supplementary valve, which is almost frictionless, and this in its turn controls the supply or discharge of water, steam, or other fluid pressure to a cylinder in which a piston works, which actuates the throttle valve or expansion gear of the engine. In this way, although absorbing a power less than half that required for one 20-candle lamp, the solenoid is able to control the most powerful expansion gear.

The Willans electric governor is shown in Fig. 89, where S is a solenoid taking the place, in incandescence lighting, of one of the lamps. In other words, the solenoid is on a branch between the main wires. The core C of the solenoid is suspended by a spring, and this spring is attached at the top to an adjusting screw used for regulating

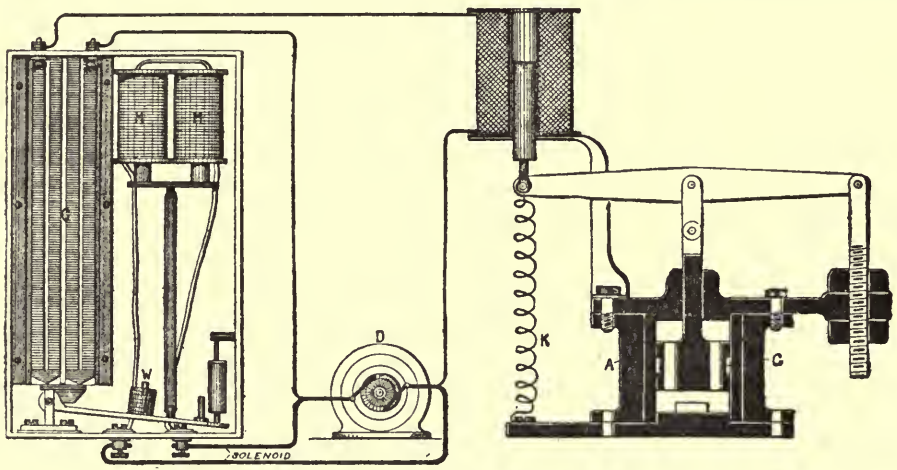


FIG. 90.—Electric Solenoid Governor.

the light. The other end of the core is connected with a small piston valve working inside the main piston W, which latter piston controls the throttle valve in the casing T. Water or other fluid pressure is admitted by the pipe P into an annular chamber surrounding the water piston W, and also, by means of a suitable passage X, into an annular space between the two small pistons which form the piston valve. The water, after actuating the piston W, escapes through the pipe E, and by means of a small piece of flexible pipe not shown.

The foregoing is a description from *Engineering* at the date ot the patent. The solenoid, however, is not sensitive enough for small variations in electric pressure. To work either this relay valve or Murray's valve, for electric governing, an electric relay is required. And for this purpose the author can recommend a carbon plate rheostatic relay, in which the resistance in a coil of a solenoid and plunger is varied by the pressure caused by a shunt current.

The best form is a carbon pile rheostat, such as that first employed by Brush on his arc dynamos, and shown in Fig. 90. There are four columns of carbon plates in series. When the turbine and dynamos are at rest the plates rest loosely on each other and on the metal blocks B on the lever L, so that the resistance is high and little current passes, in the solenoid, and the core of the solenoid is at its lowest point, the valve being full

open for the water. But if we start the machine it increases in speed and the magnet puts on pressure on the carbons, and the solenoid pulls upon the valve lever and begins to close the water inlet, so that a fixed speed is soon attained where the forces balance. The

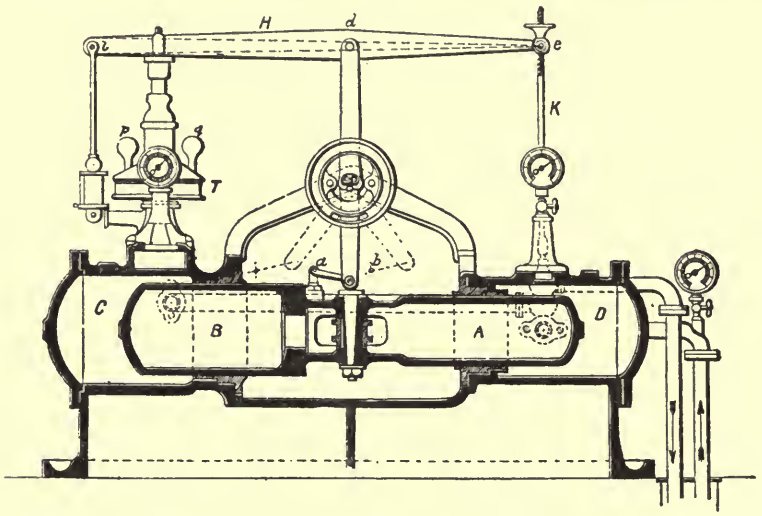


FIG. 91.—Centrifugal Governor. Sectional Elevation.

weights on the lever and the tension on the spring K are so adjusted that this resistance comes into play near the fixed speed. This device gives power with sensitiveness; for when properly adjusted the slight variation in pressure on the carbons causes a large difference in current in the regulating solenoid.

In the diagram the regulator solenoid is in series with the carbon resistance, and the pulling magnet M in parallel; but it is more sensitive when all three are in series — the solenoid, the magnet, and the resistances. It is shown connected to a Murray valve, but may be used with other valves, moving freely.

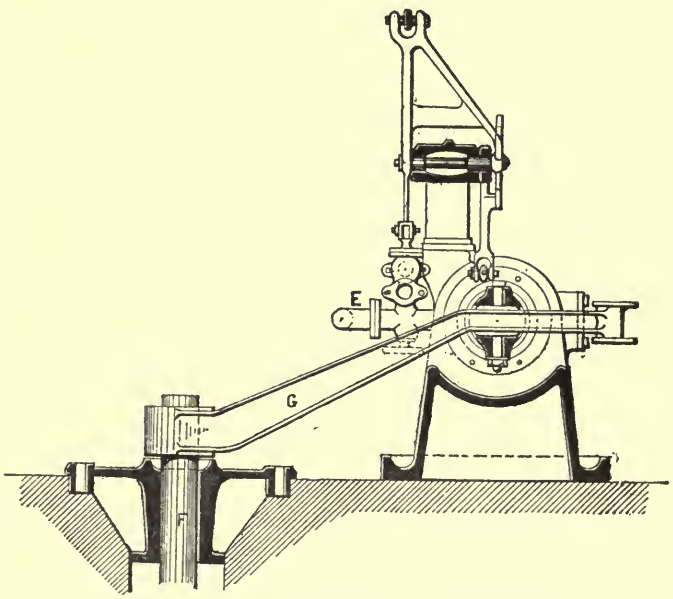


FIG. 92.—Centrifugal Governor. Cross Section.

For controlling the large inward flow turbines, governed by movable guide blades like that shown in Messrs. Gunther's diagram (p. 63), wherein the ring carrying the guide blades has to be turned through a few degrees to open or close the guide passages.

M. Rateau, in his *Turbo Machinery*, describes the combination of sensitive powerful centrifugal governor, with a differential hydraulic ram for working the guide blades. Fig. 91 is a longitudinal section, Fig. 92 a cross section. The ram is

jointed to the long lever G on the controlling shaft of the turbine. The cylinder D is always connected to the water pressure, and C is put into connection with the pressure or exhaust by the action of the governor. When connected to the pressure the ram moves to the right, the ram in C being the greater in area. When connected to the exhaust the ram moves to the left.

The valve operated by the governor is shown in Fig. 93. It is fixed to the inlet of cylinder D. A regulating needle AB is placed to regulate the exhaust outflow. The valve is worked by K from the lever H attached to the governor under T.

As the lower orifice of the cylinder D is connected to the water under pressure, while the upper orifice is in connection with the exhaust to the atmosphere, and the space between these two orifices is in communication with the large cylinder C, we see that when the index of the governor rises and the valve S is lowered the upper orifice is closed and the water under pressure flows into the cylinder C by the lower orifice; and when, on the other hand, the index descends, the valve S rises and opens fully the upper orifice, which has a section almost double that of the lower, so that the pressure falls in C and the water escapes from it. If, now, the valve is midway between these two positions, so that the two orifices present the same section to the water, the pressure in the cylinder C remains about midway between the initial pressure of the water, and the pressures on the differential piston AB are balanced so that it does not move. There is, however, this disadvantage, that there is an almost continual flow of water through *m*. On the other hand, there are two great advantages: first, the small valve S is only rarely forced to the end of its stroke by the governor; generally it is only partially moved, and from this results a speed of gate opening or closing more or less great. The point of rotation *d* of the lever H is not really fixed; it is connected to the piston valve in such a manner that it rises or falls proportionally to the motion of this piston. It results from this, that when the piston is at rest, and consequently the valve S occupies its mean position as well as the point *e*, the index *i* of the governor is obliged to fix itself in the position corresponding to that of the point *d*, and, reciprocally, the point *d* and the piston valve are obliged to follow the movement of the index.

This device of making the fulcrum rise and fall with the movement of the governor is an old one, and was used in the Willans' governor in 1884, and we have already explained that it is used on the governor in Fig. 88 (p. 71).

On the whole question of water turbine governors it may be taken that all the best makers have adopted effective hydraulic relay governing, and that close governing is obtained under all loads; but the load should not be varied suddenly to any large extent,—not because the governor would not act, but to avoid the effects of the inertia of the flowing water in the pipes and valves.

In order to prevent the bursting of the pipes when the flow is suddenly checked by the governor, ample relief valves are often provided. These are shown distinctly in Messrs. Gunther's illustrations.

Some turbines, notably the inward flow types, are more stable than others, due to the flow being opposite in direction to the centrifugal force on the whirling water.

In cases where water is not abundant and sometimes small in quantity it is better to use several turbines, and to run the number sufficient to use the available water at any time, than to use one large turbine and work it at $\frac{1}{4}$, $\frac{1}{2}$, or $\frac{3}{4}$, or whole gate; but where economy of water is no object the large turbine is cheaper in first cost and maintenance.

Other makers of governors exist, but it is difficult to get exact information of their design; but all of them working with centrifugal governors are on the relay principles, as described herein.

Gunther's turbines are operated by separate governor valve and piston, same as Murray's governor. This arrangement is shown in Messrs. Gunther's improved Pelton wheel turbine or high pressure impulse wheel, shown in Fig. 94.

That their governor is a high-class one may be seen from the following test results at an installation with a Gunther turbine and governor :—

Turbine at Landore Hotel, Keswick.—This is a Girard turbine fitted with a Gunther governor as described. Fall, 300 feet; horse-power, 15; wheel, 15″ internal diameter; speed, 850 revolutions; coupled direct to multipolar dynamo of 75 amperes at 130 volts. Used for lighting the hotel and charging an electric launch running on Derwentwater Lake.

The lights are driven direct without cells, and the cables run down to the pier and charge direct into the cells on board the launch as required. The following are the results of actual trials with the governor:—No perceptible variation of speed was noticeable on the tachometer with *sudden* load changes of 10 per cent.; with a sudden throwing off of one-third the total load by throwing out a switch, the speed rose temporarily only about 20 revolutions in 850 revolutions, equal to 2½ per cent., and then came back to normal. The turbine is usually left running with the governor all through the night, the turbine house being locked up at 11.30 p.m., and the attendant coming back at about 7.30 a.m.

This plant is shown from a photo in Plate II., wherein the governor exterior can be seen.

Much larger turbines of the same design are shown. In Plate III. is shown a view of high pressure type of impulse turbine with single jet, the illustration being from a photo of one of a set of turbines recently made for the Indian Government for utilising the Karteri Falls in the Nilghiri Hills, the power being transmitted electrically to the Coonoor Cordite Factory.

The plant comprises four of these turbines, each capable of driving 250 B.H.P. with a fall of 620 feet, and two similar but smaller turbines of 37 B.H.P. each, for driving the exciter dynamos. The large turbines are coupled direct to three phase generators of 5500 volts, and the smaller ones are also coupled direct, flexible couplings being used to couple the turbines and dynamos in each instance; and fly-wheels are also placed on the turbine shafts to obtain the utmost steadiness under load changes.

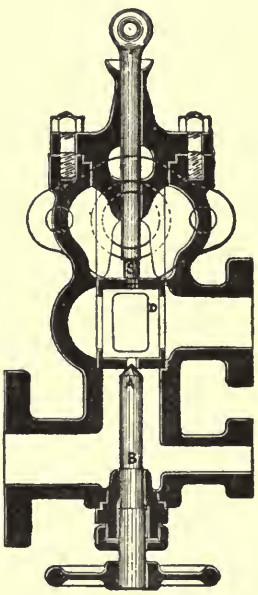


FIG. 93.—Valve of Centrifugal Governor.

In Fig. 94 we give a diagrammatic outline of the turbine, showing the port and wheel vanes in section. It will be seen that the water enters on the outer circumference of the wheel, and that the discharge is towards the centre, but in reality the greater part of the discharge is at the sides. As in the other view, the buckets or vanes are cupshaped, so that the water discharges partly towards the centre and partly at the sides. The wheel vanes are cast in bronze in a continuous ring, which is bolted to a central cast-iron plate. The guide port is provided with a sliding hood or shed, by which the area of the discharge is varied according to the power required. The lower part of the shield where it cuts off the water being properly formed so as to make it a prolongation of the guide port and preserve the true shape of the issuing jet, so that the efficiency is practically the same whether the port is full or only part open.

The four large wheels are 54 inches diameter and make 400 revolutions per minute, and the two small ones 27 inches diameter and make 800 revolutions per minute.

Each turbine is provided with Messrs. W. Gunther & Sons' hydraulic governors, and also with relief valve to minimise the danger from concussion in the pipes when load is suddenly thrown off and the supply of water checked by the closing or partial closing of the regulating hood. In this hydraulic governor the rise and fall of the governor

balls opens or closes a small valve which admits pressure water taken from the pipe main to one side or other of a hydraulic cylinder, and thus moves the piston up or down,

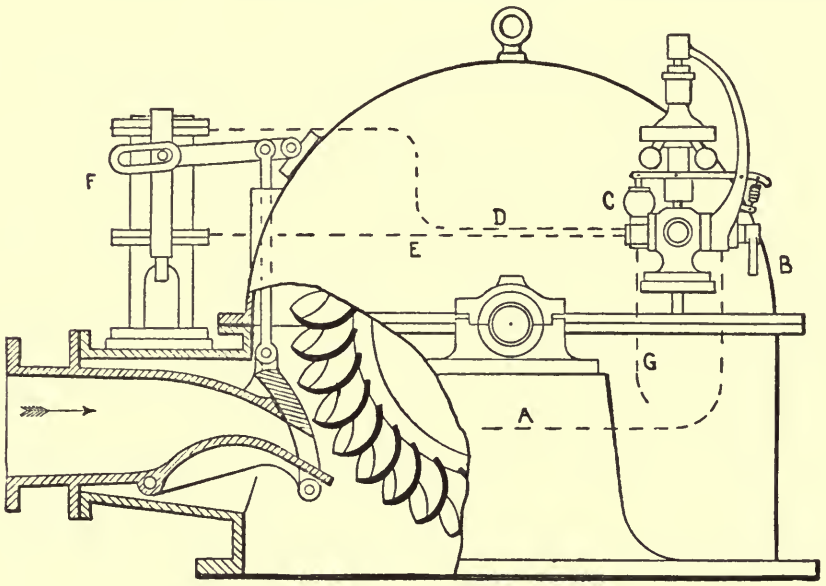


FIG. 94.—High Pressure Impulse Wheel and Governor.

and the motion of the piston is communicated by suitable lever or other connections to the regulating hood, closing or opening the jet as load is thrown off or on. The action of this governor is instantaneous, and brings the speed back to normal at all loads, that is, the speed of the turbine is kept the same at all loads between no load and full load.

FIG. 95 —Watson, Laidlaw, & Co.'s Governor.

Referring to the diagram Fig. 94, A is the pressure pipe, which takes the water from the main from a branch on the turbine inlet, a filter being affixed to the branch so that the water is filtered to remove injurious particles before it passes to the governor. The pipe A (shown as a dotted line) conducts the water through a special valve (which is used also for stopping and starting the turbine hydraulically by means of the handle B). From this valve the water passes through a piston valve contained in the valve box C, which piston valve is worked from the governor balls, and distributes the water to one or the other end of the cylinder F through the pipes D or E. By means of the valve at B the turbine can be started and stopped ; or if desired, the governor can be thrown out of action and the turbine kept running at any desired fixed gate opening.

In considering water wheel governors we have described the American forms of Pelton wheel governors. Recently a much improved automatic governor, by Messrs. Watson, Laidlaw, & Co., Glasgow, has been brought out, and is described in *Engineering* as follows:—

The wheel casing with the governor in position is shown in Fig. 95, whilst Fig. 96 represents a section through the jet nozzle.

The rod on core A is extended behind, and enlarged in diameter, to form the plunger B, which fits closely into a liner within the body of the nozzle. The helical spring inside the plunger exerts a constant pressure upon it, and tends to thrust the rod forward into the water jet, while at the same time the pressure of the water which enters at D tends to push the plunger back with greater force. The small holes E in the end of the plunger permit a certain quantity of water to pass into the chamber behind the plunger; and if the valve F, which communicates with the exhaust chamber G, be closed, the pressure behind the plunger will rise until it equals that in front, leaving the spring free to act without opposition. It is therefore apparent that the full closing of the valve F has the effect of reducing the water gate to its minimum by allowing the spring to act freely; while the contrary effect will result from the full opening of the valve, owing to the effect of the water pressure on the plunger being greater than the power of the spring.

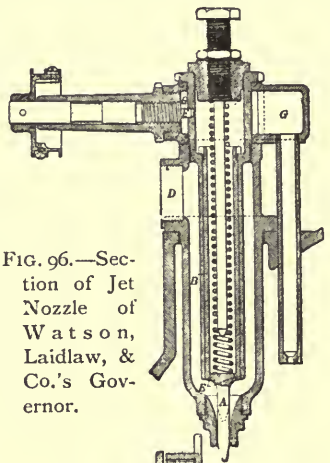
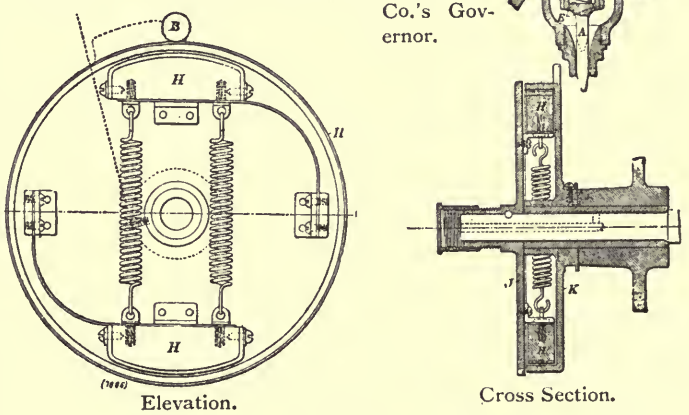


FIG. 96.—Section of Jet Nozzle of Watson, Laidlaw, & Co.'s Governor.

Partial opening of the valve F will produce water gates intermediate between the maximum and minimum, owing to the tapered form of the rod or core A. Full control of the wheel is therefore effected by the closing and opening of the valve F being made dependent upon the higher or lower speed at which the wheel runs. This result is attained through the operation of the governor shown on Fig. 97. The weights or pendulums H are attached elastically to the disc J, which is fixed to the wheel spindle and revolves with it. They are held together by springs of suitable strength, to keep them in position till a speed above normal has been reached. Thereupon they fly out, engaging on their leather-covered surfaces with the inner rim of the loose pulley K, which they tend to pull round with them. The boss of this pulley has, however, a chain wound round it, attached to the spring, and the friction of the pendulums H has to overcome the tension of this spring before they can cause the loose pulley to turn. The loose pulley is prevented from revolving through more than one revolution by the pin L; but at all speeds materially over normal the action of the pendulum will tend to draw the pulley over through an arc of a circle corresponding to the exact amount of the excess speed. It only remains to communicate this action to the spindle of the valve F in order to make the size of the water jet dependent upon the speed of the wheel. This is done by leading the chain over a wheel on the spindle of the valve, as shown in Fig. 95.



Elevation. Cross Section.

FIG. 97.—Watson, Laidlaw, & Co.'s Governor.

The governor is very sensitive, and can be adjusted to control the speed of the wheel within any required degree of variation of speed under great and sudden changes of load. It will be seen that this governor can be made to govern in two distinct ways. It can be so set as to govern the jet by regulating the flow of water proportionally to the load, or it can govern by opening the jet full and closing it quite shut, so that the time

during which water plays on the wheel is proportionate to the load. In either case the total amount of water discharged from the nozzle is proportionate to the work done.

As the Pelton wheel is specially suitable for high speeds and falls and in electrical engineering, an automatic, sensitive, and efficient governor is of utmost importance; but the difficulties which might arise from the hydraulic ram effect of opening and closing the jet entirely and quickly have not been referred to, although some means of meeting will be required.

Turbine construction in Great Britain is fairly well represented by these examples, and in recent years it has received a very considerable impulse from the perfection of electric transmission of power, enabling the power to be carried to the works at a more suitable place than where the waterfall exists.

It is to the engineer not much short of a public wrong that the greed of landowners should prohibit in many cases the use of large powers given freely and gratis by Nature in the shape of waterfalls.

A source of power running to waste is nowadays a wilful waste, and many such exist where the power could be obtained and utilised at a distance without in any way damaging the land or the amenities of the neighbourhood.

The case was different when the mill or factory had to be built adjoining the fall; whereas now a distance up to 5 miles is not an obstacle to the use of the power.

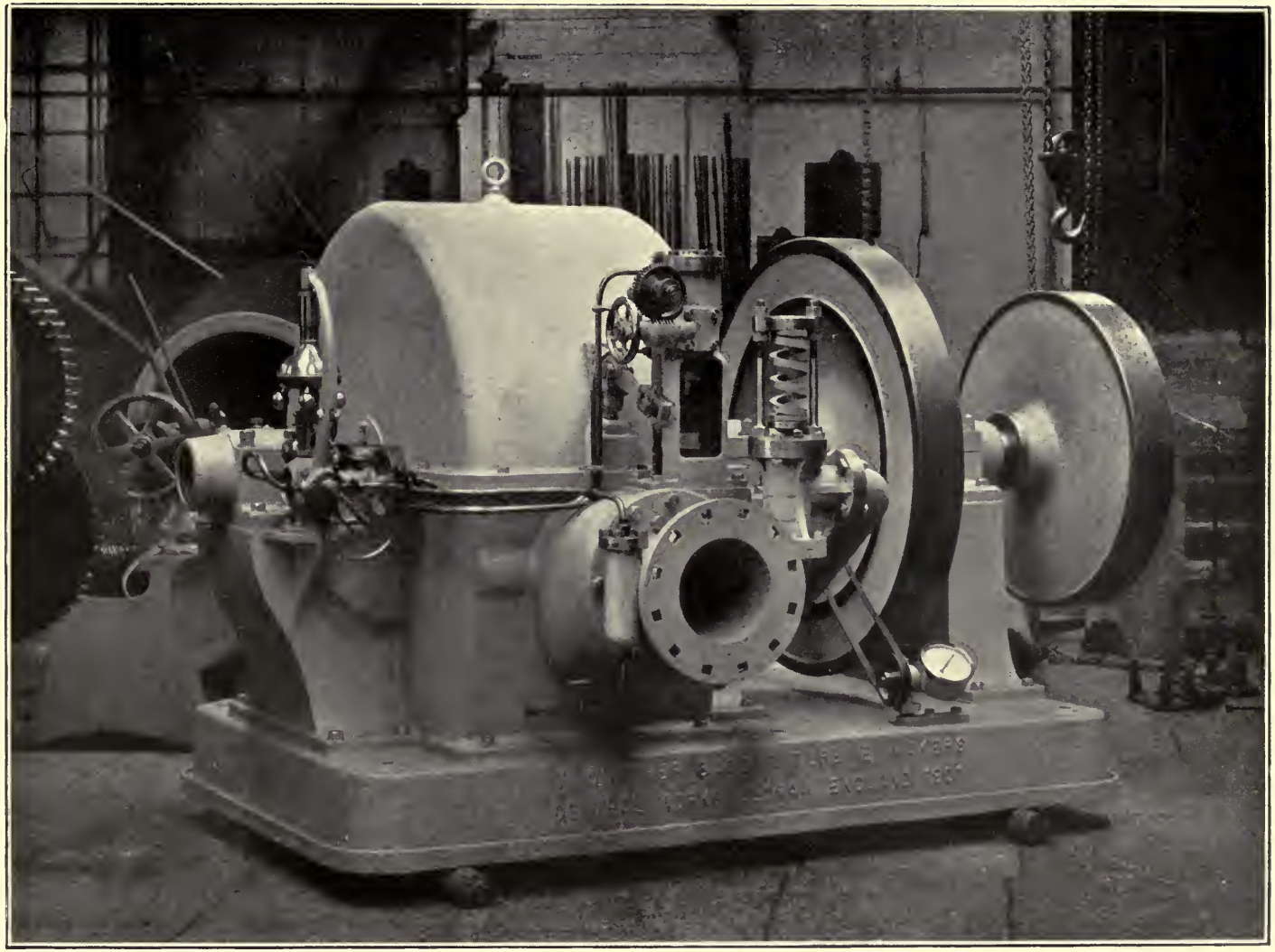
And again, in hilly countries—Ireland, Scotland, North-West England, and parts of Wales—waterfalls can be produced by the engineer, in many cases at a cost well worth the outlay, if the land could be obtained at its true value. In most instances, however, any such proposals have the instant effect of converting the land from a worthless moor or bog into most valuable property at a value prohibitive. A high fall with plenty of water is a valuable gift of Nature to the landowner, and so is the side of a hill where an engineer can form a reservoir and catch water. But as things are, they are of little use or value to the human race or the inhabitants of the country.

FLUID-ON-FLUID PRESSURE ENGINES

These might be classed among pumps, as they are usually employed for lifting water, but we wish to keep the hydraulic and pneumatic machinery in one book.

Many machines are retained in use by engineers although they are well known to be inefficient; this is only tolerable on account of great simplicity, reliability, or portability, rendering them preferable under the circumstances to a more efficiently complex or cumbrous machine. A piston pump properly designed is highly efficient, but it has many delicate parts—packing glands, shafts, fly-wheel, cranks, and rods—all accurately fitted and requiring care and attention, and only possible of construction in a well-equipped machine shop. Water may be moved up to a higher level from a well or mine by a waterfall, or by steam, by very simple but inefficient means. Thus in Hungarian mines long ago water was pumped out of them to keep them dry by means of a fall of water working a simple arrangement of pipes and vessels, such as could be readily obtained and set up by a country blacksmith.

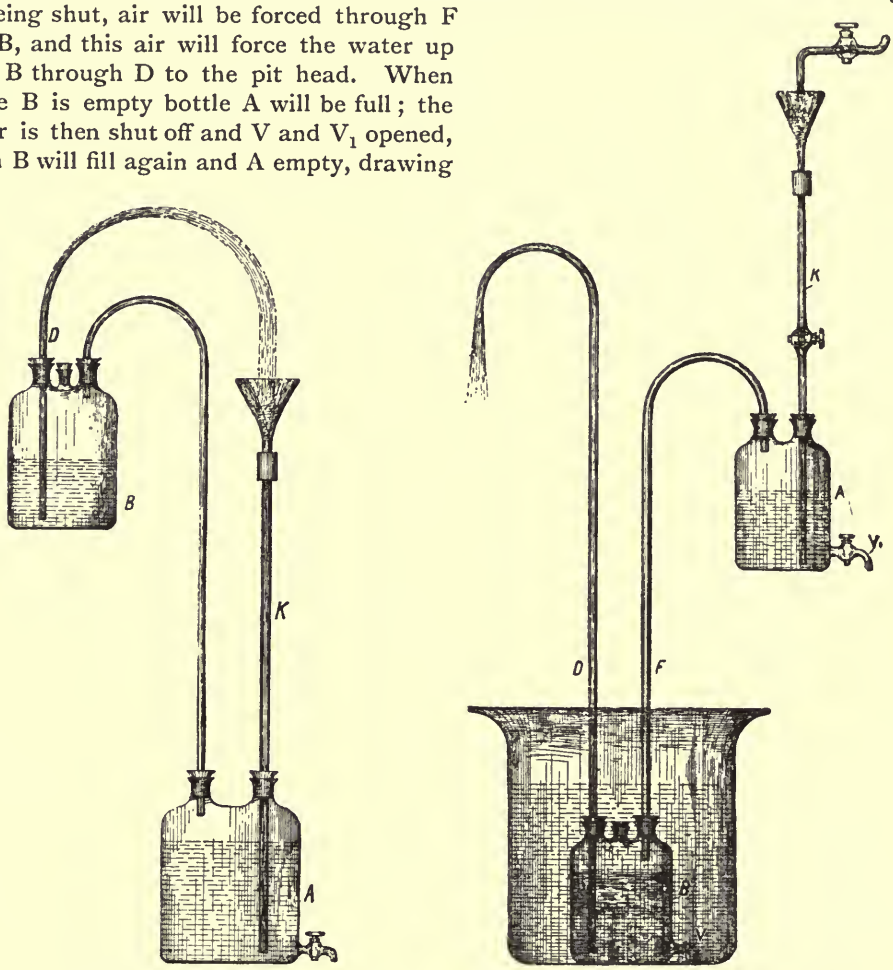
The Hungarian machine was described by Professor Rankine, who investigated it fully mathematically. It is based on the principle propounded by Hero of steam turbine fame in the second century B.C., and is the progenitor of many modern engineering apparatus. From Hero's description the apparatus was somewhat like the diagram Fig. 98, from which a working model may be made with two Wolffs bottles, three tubes, and a funnel, and five or six rubber corks. To show the experiment, bottle B is filled with water, the discharge pipe D goes to the bottom of this bottle; F, the air pipe, just enters bottles A and B; K, the supply pipe, goes to the bottom of A. To start the action, a little water is filled into the funnel, this water entering vessel A displaces the air through pipe F into vessel B, and thus presses upon the water in B and forces it up the delivery



pipe D, which pipe may be inclined so that the water falls into the funnel and maintains the action until all the water is discharged from B. In this experiment the water is raised from vessel B to a higher level.

Vessels A and B may be placed on different levels; they may be on the same level, by shortening pipe F and keeping pipe K long, as it is.

In the Hungarian machine (Fig. 98), vessel A is at the head of the pit to be drained, B is at the bottom. In this experiment the cork V is withdrawn, and B being in a tub of water, it fills with water; now shut V and allow water to fall into the funnel through K. V_1 being shut, air will be forced through F into B, and this air will force the water up from B through D to the pit head. When bottle B is empty bottle A will be full; the water is then shut off and V and V_1 opened, when B will fill again and A empty, drawing



Hero's Experiment. Experimental Hungarian Machine.

FIG. 98.—Hungarian Machines.

in air. V and V_1 are then shut, the water turned on again, and another bottle full of water pumped up, and so on until the level of the water is reduced.

Professor Rankine's description is as follows, referring to Fig. 99:—

A is the sumpt at the bottom of the mine into which the water collects. B is a receiver with a non-return valve C. D is the delivery pipe from the bottom of the receiver; there should be a foot valve. G, the waste air cock, at the top of E. The discharge valve at the bottom of E is for discharging the water which has performed its work in the working barrel. K, the supply pipe, connecting reservoir with the bottom of the working barrel E. There is an admission valve near the bottom of the supply pipe.

The valve may be opened and shut by floats in the working barrel, or by a small

auxiliary water pressure engine, or by a small wheel driven by the water discharged. A single piston valve is best, as shown in Figure.

The machine is set to work by opening the air waste cock G, the supply valve at the same time being shut. The water from the well A opens the clack C, enters and fills the working barrel B, and drives out the air through G, so that E and F only remain filled with air. Then G is shut, and remains shut while the machine is working; the discharge valve is shut and the supply from K to E opened, and the working proceeds as follows:—

The driving water descends through K into E, and compresses the air contained in E and F. The pressure so exerted on that air is transmitted to the water in B, and causes it to rise in the delivery pipe D. When the pressure has become equal to that of the column of water in D added to its resistance, the lifted water issues from D into the drain, and continues to do so until E is filled with water. Then by the valve gearing the supply valve is shut and the discharge opened; and the water in E is made to flow out, partly by its own weight and partly by the pressure of the expanding air. As soon as the air has fallen to its original pressure more water from the well flows through C into B, and drives all the air back into F and E. Then the discharge valve is shut and the supply opened, and the cycle of operations recommences.

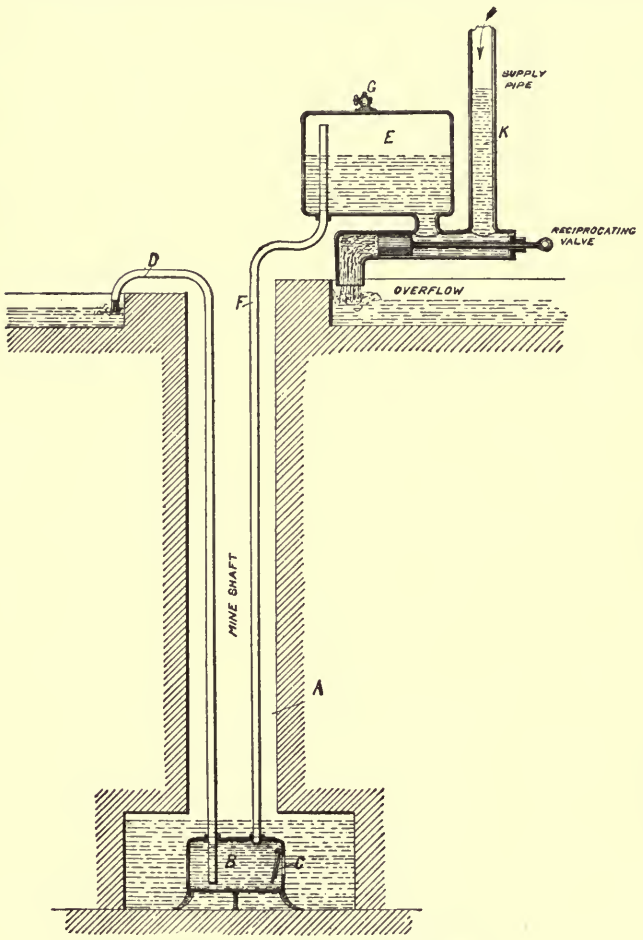


FIG. 99.—Hungarian Machine.

Let h_0 denote the head of water which is equivalent to 1 atmosphere, or 33.9 feet on an average.

Let h_1 be the height of the outlet of the delivery pipe D above the surface of the water in A; D, the weight of a cubic foot of water, or 62.4 lbs.; Q_1, the number of cubic feet per second to be raised; then DQ_1h_1 is the useful work per second.

Let h_1 be the head lost by the resistance in the pipe D, $h_0+h_1+h_2$ is the head of water equivalent to the pressure to which the air must be compressed in E, F, and B before the water will issue from the outlet of D. That pressure, in atmospheres, may be expressed thus: $n=1+\frac{h_1+h_0}{h_0}$, and the working pressure which the barrels and air pipe must be adapted to bear is $n-1$ atmospheres.

The volume of air which must pass per second from E into B, while the water is being forced out of B, is Q_1 cubic feet at the pressure of n atmospheres.

Therefore, as the original pressure of the air, before being compressed by the

descent of the water into E, is 1 atmosphere, the volume of the mass of air which descends per second, at the original pressure, is $Q = nQ_1$; and this also is the volume of water which must descend from the source per second in order to perform the work.

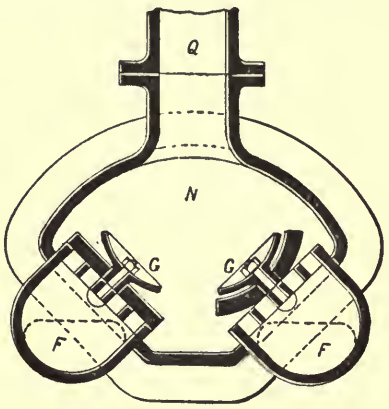
Let B and E be taken respectively to represent the capacities of those portions of the pump barrel and working barrel which are alternately filled and emptied of water at each stroke, and let F denote the capacity of the air pipe; then we must evidently have $\frac{E+F}{B+F} = n$. Let h_3 be the loss of head by the resistance of the supply pipe, valves, etc. Then the total head required for the fall is $H = h_1 + h_2 + h_3$, so that the total energy expended per second is $DQH = nDQ_1(h_1 + h_2 + h_3)$

$$= \frac{h_0 + h_1 + h_2}{h_0} \cdot DQ_1(h_1 + h_2 + h_3).$$

Comparing this with DQh, the efficiency is found—

$$\frac{Q_1 h_1}{QH} = \frac{h_1}{n(h_1 + h_2 + h_3)} = \frac{h_0 h_1}{(h_0 + h_1 + h_2) \cdot (h_1 + h_2 + h_3)}.$$

The reduction of efficiency by the factor $\frac{1}{n}$ in the above, corresponding to loss of head equal to $\left(1 - \frac{1}{n}\right)H$, is the loss of energy due to compressing the air and water friction.



Valves of Pulsometer.

We have already referred to the Thomson jet pump (p. 40). We may now refer to the pulsometer pump, which is also a fluid-on-fluid pump; steam being directly applied to the water to force it out, a vacuum is formed by the condensation of the steam, which by suction draws the water up into the pump. In this place we will only refer to it briefly to show its principles of action; its construction falls under the head of Pumps.

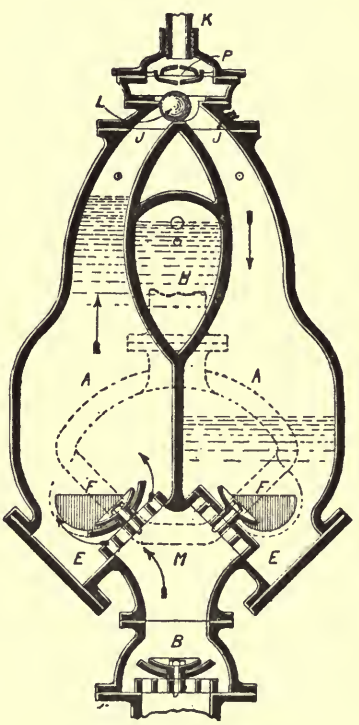


Fig. 100.—Pulsometer.

Referring to Fig. 100, there are two side chambers AA to receive the water alternately, and an intermediate vessel H, whose purpose will be explained. EE are suction and GG delivery valves (Fig. 100), B a foot valve, N the delivery chamber connected to A by short pipes FF, and Q the rising main or delivery pipe. To start the pump, the three vessels are filled through the hole C, the water resting on foot valve B. The ball L being compelled to lie on one or the other seat JJ, steam is admitted at K, and, entering, say, the right-hand passage, displaces the water through F, until the level falls to the upper edge of the orifice. Steam then blows through into F with some violence, causing a partial vacuum in A. The ball being now drawn to the right-hand seat, water rises into the right chamber ready for the next stroke, steam enters the left chamber, and the action is continuously repeated. The vessel H serves the purpose of an air vessel, to steady the flow into N; and to prevent the sudden shock caused by the rush of suction water, air cocks DD are placed on the three vessels, and kept open. The "Grel" valve (Fig. 101) at P is applied to economise the steam supply. It is a short hollow piston,

which rises and falls on account of the difference of pressure within and without it, thus closing the pipe K after a portion of the stroke has been completed; the remainder of the stroke being completed by steam expansion.

It is not economical by any means, but it is nevertheless a very useful and, in many cases, indispensable machine.

The hydraulic ram next claims

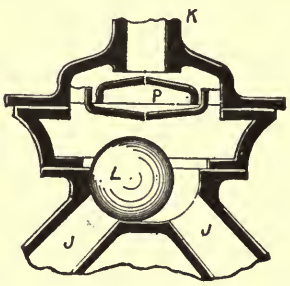


FIG. 101.—Grell Valve.

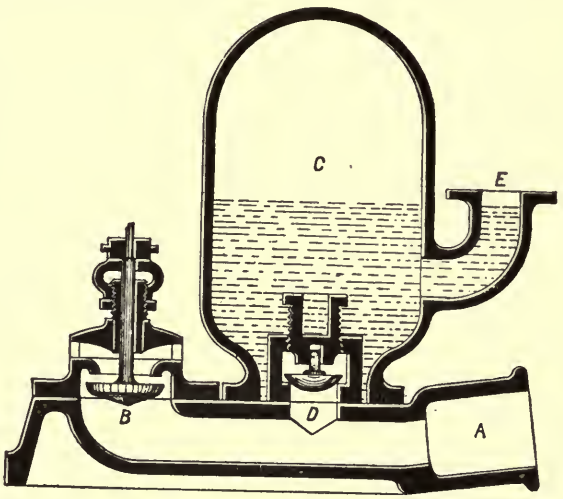


FIG. 102.—Hydraulic Ram.

our attention. Here we have a machine or engine not only simple, but highly efficient, and a valuable means for providing a high pressure water supply from a low fall of water.

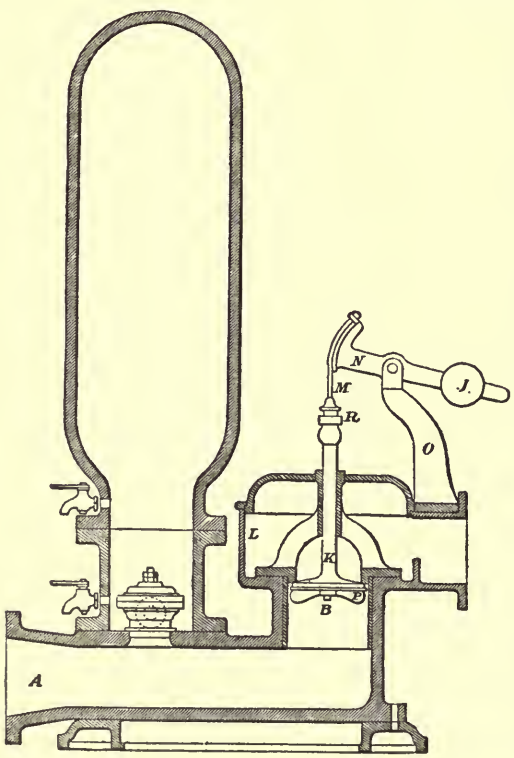


FIG. 103.—Balanced Pulse Valve.

Hydraulic rams are divided into two classes. In the first, the water of the fall is employed to raise a portion of itself to a higher level; in the second, the fall is used to raise some other water or fluid to a higher level, and hence they are called pumping rams.

The principles of action are simple. If water is allowed to flow freely through a long pipe it acquires a certain velocity, due to the difference in height between the upper and lower end, and if the lower end is suddenly closed the whole column of moving water in the pipe, in stopping, delivers up its momentum by pressing violently against the stop and the sides of the pipe; and if we had a small valve on the side of the pipe near the stop, the water would be forced out with considerable pressure in a sudden spurt.

It is thus only necessary to have a long pipe, with one end in the water supply and the other end some feet below the water level, with a stopper valve and a delivery valve; an air vessel is added in order to regulate the pressure. The stopper valve is called the pulse valve, because it beats regularly when the pump is in action. The construction may be gathered from Fig. 102.

Into socket A the supply pipe fits; this pipe is called the drive pipe. B is the pulse

valve and D the delivery valve; C is the air vessel and E the rising main. It will be obvious that a great deal depends upon the weights and sizes of the valves, especially of the pulse valve B. If it is too heavy it will not close when the water acquires its full velocity in flowing out of it; if it is too light, then it will close before the water has acquired the full velocity due to the fall. Hydraulic rams made on this simple plan are therefore not easily adapted to different falls: the pulse valve suitable for one fall not being correct for another fall.

Again, in order to get the full force of the fall the pulse valve must have a large opening to allow the water free exit, so that it becomes heavy. It is therefore clear that this valve should be balanced by an adjustable spring or weight, so that it can readily be adjusted to obtain the best effects under all conditions of working. This improvement was made by Mr. Keith, whose machine is shown in Fig. 103. The valve B is hung on a lever N and balanced by a weight J, so that it can be nicely adjusted to open and close at the right time on any fall.

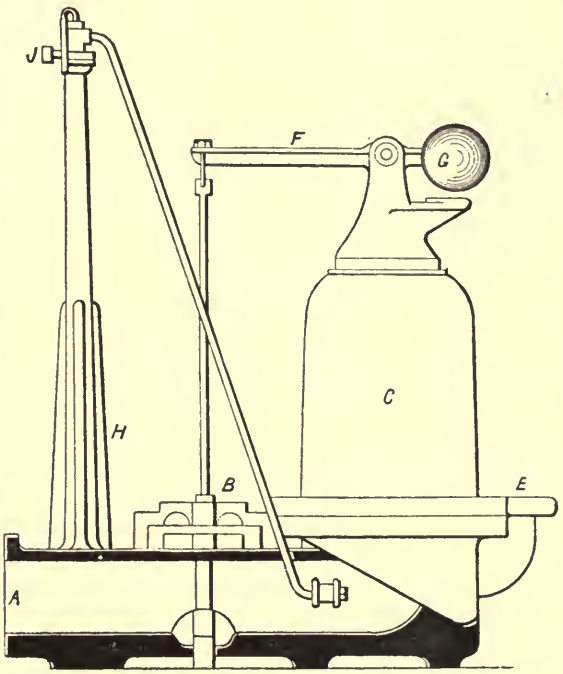


Fig. 104.—Balanced Pulse Valve.

Another balanced ram is shown in Fig. 104, an American design. The pulse valve is a piston form guided at the lower end in a socket, which has a rubber stop to silence the beats; it is balanced by weight G on lever F.

In this ram there is a device for maintaining air in the air vessel. Air is absorbed by water under pressure, and hence in time it diminishes in the air vessel, and some means for replenishing it has to be provided. Small valves called snifting valves have been adopted, but they are not always effective. The air is in the vessel under pressure, and to get any more air into it force must be used. In this ram a long, small bore pipe H is connected to the drive pipe A; it has a valve J at the top which allows air to enter, and which closes when the pressure comes on; the air is then compressed and forced into the air vessel, a small quantity at each pulse.

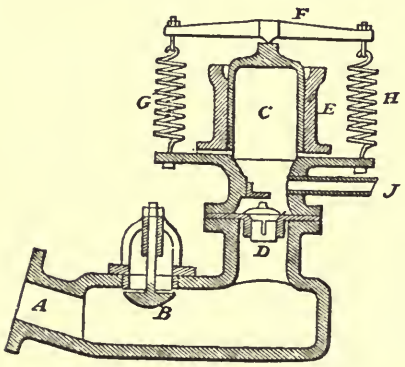


Fig. 105.—Spring-Balanced Ram.

In Blake's hydraulic ram (Fig. 105) springs were used instead of air. C is a piston, sliding water-tight in a cylinder and held down by a lever and springs; the water entering C raises it up at every pulse, and thus acts regularly.

Another device, by Messrs. Easton & Co., whereby air is compressed in a chamber in connection with the drive pipe and some of it carried into the air vessel, is shown in Fig. 106. When the pulse valve is open, and the water flows past the opening F, air fills the small direct connected vessel C through a snifting valve D. Upon the valve B

closing, some of the water forces itself into C and some into the smaller vessel F; thus the violence of the compression is reduced by the air spring in F, which forces the water back again through D into C with air absorbed during the high pressure.

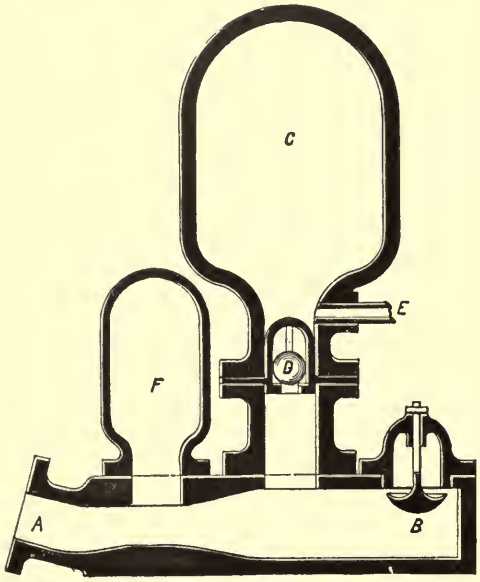


FIG. 106.—Double Air Vessel Ram.

We shall only refer to one form of pumping ram. This machine is used for raising clean water from a well by means of a fall from a river or drain of unclean water. It has some useful applications, but is no longer a simple machine. It must be remembered that these hydraulic machines are not economical, and only on account of their simplicity can they be recommended. When we therefore depart from a simple construction and introduce pistons, rods, valves, levers, and make a complicated mechanism of it, the advantages disappear, and we may well consider whether a turbine and pump would not in the circumstances be better for the purpose.

Blake's pumping ram is shown in Fig. 107. In this pump a piston C is driven up by the recoil of the drive water. On the other end of the piston rod is a pumping piston D; F is the suction pipe into the well; H, the delivery water valve into receiver K, and J is the rising main. The spring forces the pistons back into their lowest position between each pulse.

In the same way air may be compressed and used in rock drills and other mining tools. And great use was made of hydraulic ram engines in compressing air for driving the tunnel through Mont Cenis, water being unlimited in quantity.

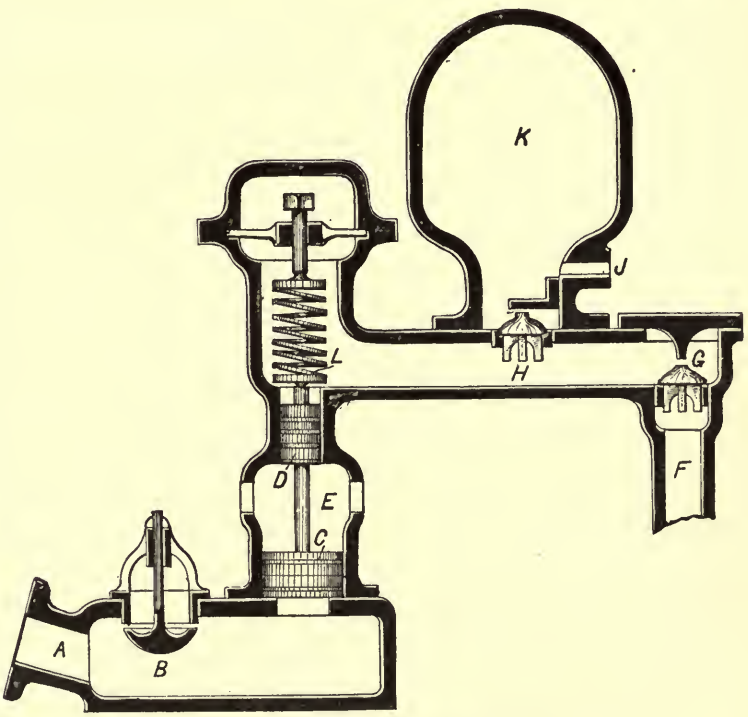


FIG. 107.—Pumping Hydraulic Ram.

In setting out a hydraulic ram it is necessary first to ascertain the quantity of water available for the drive, and to survey the ground to obtain the position for the ram where it may obtain the greatest fall through a pipe of at least 30 feet long. It

is evident that a long pipe will be required for a high lift, for the force at the moment of closing the pulse valve is proportional to the mass of water in motion in the drive pipe and to its rate of motion. On the other hand, it is useless to make the pipe too long; for while a long pipe gives a greater pressure, it beats slower than a short pipe. In practice the length of the drive pipe is limited often by the nature of the fall and the location of the ram, which has to be selected for the running off of the tail or waste water. The shortest length is about five times the height of the fall, and where possible it may be increased to ten times the height of the fall.

The table here given is one compiled by the *American Engineer*, gives the coefficient for various quantities of available drive water, which, multiplied by the coefficient for any given height to which the water is to be forced, will give the quantity delivered.

Thus if we had a 20-foot fall to deliver 50 feet high, the quantity delivered would be, for 100 gallons available per minute, $0.3282 \times 100 = 32.82$ gallons delivered per minute.

TABLE VI.—ELEVATION.

Working Head.	15	18	21	24	27	30	35	40	45	50	60	70	80	90	100
Feet.															
2	.0724	.0533	.0420	.0307	.0235	.0181	.0112	.0063	.0027	...	...	...	...	...	...
3	.1327	.1020	.0807	.0651	.0530	.0441	.0326	.0243	.0181	.0132	.0063	.0017	...	...	...
4	.1960	.1535	.1234	.1020	.0854	.0724	.0560	.0441	.0348	.0281	.0180	.0112	.0063	.0027	...
5	.2614	.2068	.1686	.1404	.1189	.1020	.0807	.0652	.0533	.0441	.0307	.0217	.0150	.0099	.0063
6	.3282	.2614	.2146	.1800	.1535	.1327	.1063	.0870	.0724	.0608	.0441	.0325	.0243	.0180	.0132
7	.3960	.3170	.2614	.2203	.1885	.1640	.1327	.1096	.0920	.0782	.0580	.0441	.0340	.0264	.0205
8	.4647	.3733	.3090	.2614	.2248	.1960	.1595	.1327	.1121	.0960	.0724	.0560	.0441	.0351	.0281
9	.5341	.4303	.3572	.3030	.2614	.2285	.1868	.1561	.1327	.1142	.0870	.0682	.0545	.0441	.0360
10	.6040	.4877	.4058	.3450	.2984	.2614	.2145	.1800	.1535	.1327	.1020	.0807	.0651	.0533	.0441
11	.6745	.5459	.4549	.3874	.3357	.2947	.2425	.2041	.1746	.1514	.1172	.0934	.0760	.0627	.0524
12	.7453	.6040	.5043	.4302	.3733	.3282	.2708	.2285	.1960	.1704	.1327	.1063	.0870	.0723	.0608
13	.8166	.6627	.5540	.4732	.4112	.3620	.2994	.2532	.2177	.1896	.1483	.1194	.0983	.0821	.0694
14	.8881	.7217	.6040	.5166	.4494	.3960	.3282	.2780	.2395	.2090	.1640	.1327	.1096	.0920	.0782
15	.9600	.7809	.6543	.5601	.4877	.4303	.3572	.3030	.2614	.2285	.1800	.1460	.1211	.1020	.0870
16	...	.8404	.7048	.6040	.5263	.4647	.3863	.3282	.2835	.2482	.1960	.1595	.1327	.1121	.0960
17	...	.9001	.7555	.6480	.5650	.4993	.4157	.3535	.3058	.2680	.2123	.1731	.1444	.1223	.1050
18	...	.9600	.8064	.6921	.6040	.5341	.4451	.3790	.3282	.2880	.2286	.1868	.1561	.1327	.1142
19	...	...	.8574	.7364	.6430	.5690	.4746	.4046	.3507	.3081	.2449	.2006	.1680	.1430	.1262
20	...	...	.9086	.7808	.6823	.6040	.5042	.4303	.3733	.3282	.2614	.2145	.1800	.1535	.1327
21	...	...	.9600	.8254	.7217	.6392	.5340	.4561	.3960	.3486	.2780	.2286	.1920	.1640	.1420
22	...	...	....	.8701	.7612	.6745	.5640	.4820	.4188	.3688	.2947	.2425	.2041	.1746	.1514
23	...	...	...	.9150	.8007	.7098	.5940	.5080	.4417	.3892	.3114	.2567	.2163	.1853	.1609
24	...	...	...	.9600	.8404	.7453	.6241	.5341	.4657	.4097	.3282	.2708	.2185	.1960	.1704

The old rule for finding the diameter of the drive pipe $D = \sqrt{(1.63Q)}$, Q being the available amount of water in cubic feet; and if h = the height to which the water is to be lifted and H the fall, then L, the length of the drive pipe, should be $L = H + h + \frac{h}{H} \times 2$.

In the case just taken, $H = 20$, $h = 50$; then $L = 20 + 50 + \frac{50}{20} \times 2$ feet $= 145$ feet.

The capacity of the air vessel should be equal to the capacity of the drive pipe, and the area of the opening of the pulse valve should be nearly the area of the drive pipe in section; per 100 gallons, D would be $= \sqrt{(1.63 \times 16)} = 5$ inches nearly.

The air vessels are usually cast iron, well pitched inside to fill any pores up through which air might escape. Glass has also been used for small pumps.

Hydraulic rams require to have well made and well designed valves, and occasional attention to see that the air is sufficient in the air vessel. In country houses they are much used in all countries to provide a household water supply under pressure.

We now may consider other fluid-on-fluid pressure engines, in which gases act upon liquids and *vice versâ*.

STEAM JETS

The air spray is familiar to every one, and illustrates the production of a partial vacuum produced at right angles to a jet of fluid. We have many interesting applications of this principle in engineering.

The acting part in these apparatus is a jet of air, steam, or water, which inducts motion in a surrounding mass of fluid. In the simple cross blast jet the horizontal jet passing across the mouth of the vertical pipe acts simply by cutting off the atmospheric pressure from that mouth, and so the liquid rises in the vertical tube and is blown away as a spray. The velocity of the jet of air, or steam, or water is sufficient to carry it across the mouth of the tube without deflection, even although there is a vacuum beneath.

In the same way the jet acts in nozzles. Here the velocity of the jet first carries air out of the annular space, making a partial vacuum into which the liquid rushes; the liquid coming in contact with the jet is carried on also by contact.

As nozzles are the important organs of these induction machines called injectors and ejectors working on these principles, we must consider nozzles, or hollow cones, in various types.

A converging cone delivers a rod of fluid without contraction to the same extent as a straight tube or through a hole in a plate.

Steam being the most important fluid at present, and more information being collected regarding it than about other gases, we will first consider the steam jet.

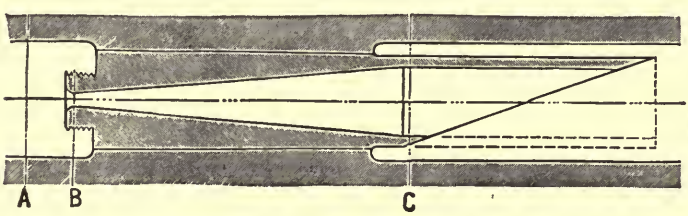


FIG. 108.—De Laval Steam Nozzle.

Experiments have been made with a conical nozzle, as shown in Fig. 108. This nozzle may be used with either end as the inlet and the other end the outlet.

With the wide end as the inlet and the narrow end the outlet, we get the fluid gradually increasing in velocity up to the narrow exit, where it issues, slightly converging at the full speed due to the height or pressure.

If we reverse the nozzle we get a very different result.

Suppose we consider the nozzle connected to a steam boiler at 200 lbs. pressure about atmospheric, or 214.7. According to Professor Zeuner, in a nozzle in which steam is adiabatically expanded the potential energy (heat) is converted into kinetic energy. The kinetic energy of 1 lb. of steam at velocity V in feet per second is of course equal to—

$$\frac{V^2}{2g} \text{ foot-lbs.} \quad (1)$$

Suppose steam is let down from pressure p_1 to pressure p_2 adiabatically in an expanding nozzle. The internal heat of the steam at p_1 = (the internal heat of the steam at p_2) + (kinetic energy at p_2 in heat units); or, in other words, the kinetic energy is proportional to $H - h$, where H is the internal heat at p_1 and h that at p_2. And if J is the mechanical equivalent of heat, the velocity of the steam acquired by the time it had fallen to p_2 would be—

$$V = \sqrt{2gJ(H-h)}. \quad (2)$$

The internal heats depend on the percentage of moisture in the steam, calculated on the assumption of constant entropy during the expansion.

Theory and experiment agree that at a certain ratio between p_1 and p_2 a maximum amount of steam flows through a converging nozzle. This limiting ratio is—

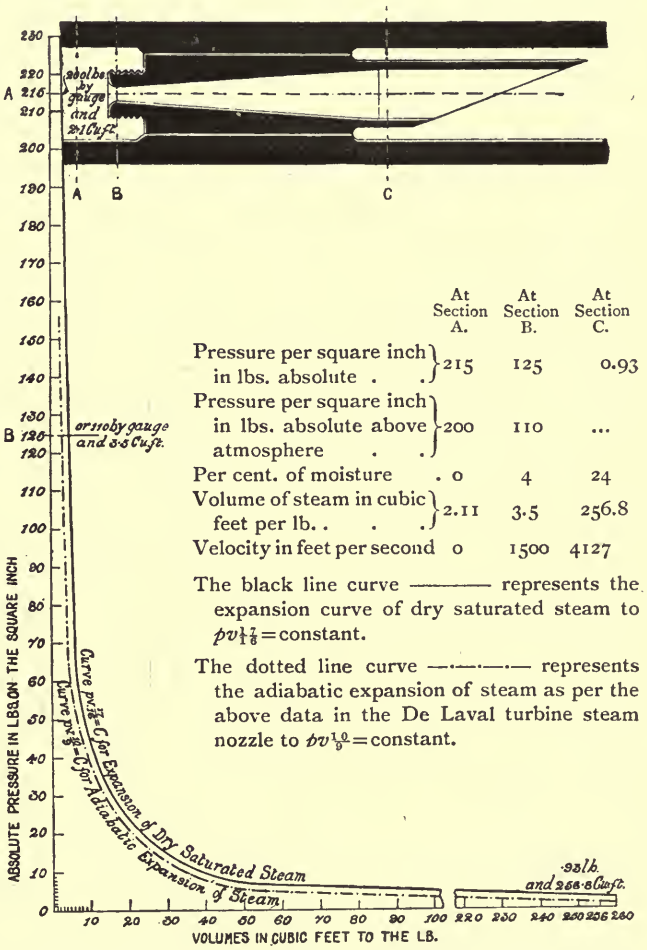
$$\frac{p_2}{p_1} = 0.577 \text{; and hence } p_2 = 0.577 \times p_1.$$

In the case cited ot a pressure p_1 equal to 214.7 absolute; p_2, that is at section B of the nozzle, the pressure would equal $0.577 \times 214.7 = 124$ lbs. (3)
Absolute whichever end of the nozzle we used as the inlet, or 109.2 gauge pressure.

The steam, therefore, which escapes from section B has still much pressure, and this can be seen by making the large end the inlet and blowing the steam into the atmosphere, when the issuing jet will be seen to instantly expand widely.

By using the nozzle at the narrow end as the inlet, the steam, after passing section B, is constrained to flow forward at a still increasing velocity as its pressure falls, until it reaches the outlet, where it cannot expand further, and issues as a cylindrical jet at the full velocity due to the pressure difference.

Professor Jamieson has plotted out the following curves (Fig. 109) of the expansion in his contribution to the discussion on the De Laval steam turbine, with a view of explaining not only the sudden fall of pressure near to and in the throat of the steam nozzle of the De Laval turbine, but also the further fall ot pressure along the conical part to its mouth. The full line curve represented the natural loss of pressure in dry saturated steam as it expanded in accordance with Professor Rankine's well-known formula $pv^{\frac{17}{16}} = a$ constant; where p was the pressure in lbs. per square inch absolute, and v the corresponding volume in cubic feet per lb. of steam. This curve is drawn from 475 lbs. per square inch—at which pressure 1 lb. of the steam occupied 1 cubic foot—down to 1 lb. absolute, at which it occupied 330 cubic feet. In the Figure is included the range of pressures specially mentioned in the paper. The dotted line represented an adiabatic expansion curve to $pv^{\frac{10}{9}} = a$ constant, from 215 lbs. absolute at Section A, before entering the nozzle, down to 0.93 lb. at Section C, where it occupied 256.8 cubic feet, and left the nozzle with a velocity of 4127 feet per second with 24 per cent. of moisture. This curve passed through the point B, where the steam occupied 3.5 cubic feet, and has 4 per cent. of moisture, with a velocity of 1500 feet per second. Now, it was evident from this curve that if the potential energy of each lb. of static steam at A had been so far converted into kinetic energy at B that it had there a velocity of 1500 feet per second, and contained 4 per cent.


A
B
C
200lbs by gauge and 2·1 Cu ft.
215 lbs by gauge and 0·5 Cu ft.
·93 lb. and 256·8 Cu ft.
ABSOLUTE PRESSURE IN LBS ON THE SQUARE INCH
VOLUMES IN CUBIC FEET TO THE LB.
Curve $pv^{\frac{17}{16}}$ = C for Expansion of Dry Saturated Steam
Curve $pv^{\frac{10}{9}}$ = C for Adiabatic Expansion of Steam

	At Section A.	At Section B.	At Section C.
Pressure per square inch in lbs. absolute	215	125	0.93
Pressure per square inch in lbs. absolute above atmosphere	200	110	...
Per cent. of moisture	0	4	24
Volume of steam in cubic feet per lb.	2.11	3.5	256.8
Velocity in feet per second	0	1500	4127

The black line curve ———— represents the expansion curve of dry saturated steam to $pv^{\frac{17}{16}} = \text{constant}$.

The dotted line curve —·—·—·— represents the adiabatic expansion of steam as per the above data in the De Laval turbine steam nozzle to $pv^{\frac{10}{9}} = \text{constant}$.

FIG. 109.—Comparison of Data and Curve from the De Laval Nozzle, with the Natural Expansion Curve for Dry Saturated Steam.

of moisture, with an increase of volume from 2.11 cubic feet at A to 3.5 cubic feet at B it must of necessity have fallen in pressure from 215 to 125 lbs. absolute pressure in doing so. This is in strict accordance with the natural law for the adiabatic expansion of steam. The temperature of the steam must also have fallen from 382° Fahr. to at least 340° Fahr. in this short passage. It might be that the steam in passing from A to B, and from the very small throat of $\frac{1}{4}$-inch diameter, at the rate of 8 lbs. weight per second, naturally formed a *vena contracta* due to "throttling." In any case, it must there lose potential energy due to friction and increased velocity. This would account for its expansion from 1 lb. of dry saturated steam at A to that of slightly moist steam at B, with a corresponding and natural loss of pressure in temperature during its increase of velocity from 0 to 1500 feet per second from A to B, and corresponding fall in pressure.

TABLE VII.—THE VELOCITY OF OUTFLOW AND THE WORKING CAPACITY OF DRY SATURATED STEAM FROM EXPANDING CONES.

KONRAD ANDERSSON.

Initial Steam Pressure, Lbs. per Square Inch.	Counter-pressure 1 Atmosphere.			Counter-pressure 2.4 Lbs. per Square Inch Absolute, corresponding to 25 Inch Vacuum.			Counter-pressure 0.93 Lbs. per Square Inch Absolute, corresponding to 28 Inch Vacuum.		
	Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy Foot-lbs. per Second.	H.P. of 550 Foot-lbs. per Second.	Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy Foot-lbs. per Second.	H.P. of 550 Foot-lbs. per Second.	Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy Foot-lbs. per Second.	H.P. of 550 Foot-lbs. per Second.
		Per Lb. of Steam per Hour.			Per Lb. of Steam per Hour.			Per Lb. of Steam per Hour.	
60	2421	25.29	0.046	3320	47.57	0.087	3680	58.44	0.106
80	2595	29.06	0.053	3423	50.56	0.092	3793	62.08	0.113
100	2717	31.86	0.058	3520	53.47	0.097	3871	64.66	0.118
120	2822	34.37	0.062	3596	55.80	0.101	3940	66.99	0.122
140	2913	36.62	0.066	3661	57.84	0.105	3999	69.01	0.125
160	2992	38.63	0.070	3718	59.65	0.108	4045	70.61	0.128
180	3058	40.35	0.073	3764	61.14	0.111	4091	72.22	0.131
200	3115	41.87	0.076	3810	62.64	0.114	4127	73.50	0.134
220	3166	43.26	0.079	3852	64.03	0.116	4159	74.64	0.136
280	3294	46.83	0.085	3962	67.74	0.123	4229	77.18	0.140

In Mr. Konrad Andersson's paper he gives an example of the action of the steam in the expanding nozzle,—the narrow end connected to boiler at 200 lbs. pressure, the wide end into a 28 inch vacuum, steam supposed to enter dry.

The general formulæ for the velocity of a gas in feet per second due to a difference of pressure is $V = \sqrt{2gh}$, where h is the height of a column of the gas whose weight balances the difference of pressure. But this formulæ has its limitations, for beyond a difference of pressure less than 58 per cent. of the initial pressure the flow is constant. The velocity of the discharge of steam from a boiler at 100 lbs. pressure is no greater into a vacuum than it is into another boiler at 58 lbs. pressure.

But this is true only for a converging nozzle discharging at the narrow end, and suddenly and unrestrainedly expanding; if, however, the nozzle is prolonged and gradually diverges, the velocity increases and becomes proportional to the difference of pressure between the narrow nozzle and the wide end. The final velocity of the steam is acquired in two stages. First, it acquires a velocity of 1500 feet per second in passing from the boiler to and through the converging parts, and a further 2627 feet in expanding in the diverging part of the nozzle, making a final velocity of 4127 feet.

TABLE VIII.—PROPERTIES OF STEAM.

Absolute Boiler Pressure per Square Foot.	Absolute Boiler Pressure per Square Inch.	Volume of 1 Lb. of Steam. Cubic Feet.	Weight of 1 Cubic Foot of Steam. Lbs.	Temperature of Steam. ° Fahr.	Velocity of Steam Jet, calculated from Narrow Nozzle. Feet per Second.	Square Root of Product of Density × Pressure per Square Foot.
144	1	330.4	.0030	102.8	1238	.6
288	2	171.9	.0058	126.3	1262	1.3
432	3	117.3	.0085	141.6	1277	1.9
576	4	89.51	.0112	153.1	1291	2.5
720	5	72.56	.0138	162.4	1299	3.1
1,440	10	37.83	.0264	193.3	1326	6.2
2,160	15	25.85	.0386	213.0	...	9.2
2,880	20	19.74	.0506	227.9	...	12.1
7,200	50	8.338	.1199	280.9	1398	29.4
10,080	70	6.076	.1645	302.7	...	40.1
11,520	80	5.358	.1866	311.8	...	44.8
12,960	90	4.796	.2085	320.1	...	52.0
14,400	100	4.342	.2302	327.6	1420	58.0
15,840	110	3.969	.2519	334.6	...	63.2
17,280	120	3.656	.2735	341.0	...	69.0
18,720	130	3.390	.2949	347.1	...	74.0
20,160	140	3.161	.3163	352.8	...	80.0
21,600	150	2.962	.3376	358.2	1434	86.0
23,040	160	2.786	.3588	363.3	...	91.0
24,480	170	2.631	.3800	368.2	...	96.0
25,920	180	2.493	.4012	372.8	...	102.2
27,360	190	2.368	.4223	377.3	...	108.0
28,800	200	2.256	.4433	381.6	1452	112.1
36,000	250	1.825	.5478	400.8	1454	140.4

Taking the three sections of the cones :—

Section A—

Pressure, 200 lbs. above atmosphere.

Moisture percentage = 0.

Unit quantity of steam, 1 lb.

Section B (smallest section)—

Pressure, 110 lbs.

Quantity of steam, 0.96. Moisture, 0.4 lbs.

Velocity $V = \sqrt{2gJ(H-h)} = 1500$ feet per second.

Volume of steam, 3.5 cubic feet.

Section C (the wider end)—

Pressure, 2 inches of mercury (absolute pressure).

Percentage of moisture in the steam, 24 per cent.

Specific quantity of steam, 0.76.

Velocity of the steam, 4127 feet per second.

Specific volume of the steam, 256.8 cubic feet per lb.

The proportion between the areas of the large and small section of this nozzle should be as 27.2345 to 1, or the proportion between the diameters of these two sections as 5.2187 to 1. If, for instance, the diameter of the small section is 6 millimetres, or very nearly $\frac{1}{4}$ of an inch, the diameter of the large section should be 31.31 millimetres, or nearly $1\frac{1}{4}$ inch. Through such a nozzle there passes a certain constant weight of dry saturated steam of 200 lbs. pressure per hour, neither more nor less. This fact of the nozzle passing only a certain amount of steam per hour is used as a measure of steam.

The quantity of steam passed by the nozzle is measured by the section B, and is equal to $Q = 370\,AD$ so long as the difference of pressure is greater than $\frac{3}{5}$ the initial

pressure. A is the area of the section at B, and D the weight of 1 cubic foot at 215 absolute pressure; hence $370 \times 0.25'' \times 0.5 = 462.5$ lbs., taking A as $\frac{1}{4}$ inch and 1 cubic foot of steam at that pressure = 0.5 lbs. weight approximately.

We have seen that the impulse per second of a fluid is equal to $\frac{W \times V}{g}$. From this we can proceed to find the impulsive power of a jet. Its kinetic energy $= \frac{V^2}{2g}$ in foot-lbs. Say it issues from an expanding cone at 4000 feet per second, then the horse-power per lb. of steam would be

$$= \frac{4000^2}{64.4 \times 550 \times 3600} = 0.125 \text{ horse-power total}$$

impulsive energy per lb. of steam.

But if the steam strikes a solid body or a liquid body, its whole kinetic energy is not obtained. In a steam turbine the steam leaves the buckets with about a third of its velocity still in it. When it strikes water, as in an injector, a large quantity of its energy merely heats the water.

INJECTORS

We now come to consider the nozzles and their action. All that is known in the actual design of injectors, ejectors, and kindred machines have been found by experiment. We can, as we have just seen, calculate the efflux of a fluid from an orifice, find its quantity and velocity if we know the impelling pressure or head; and if we know this and want to find the area of an orifice to deliver a certain quantity

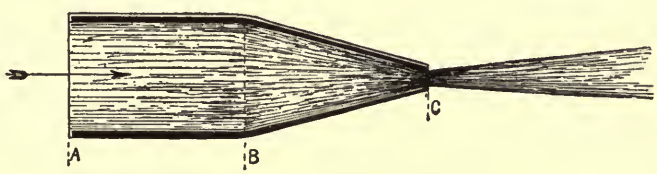


FIG. 110.—Simple Nozzle.

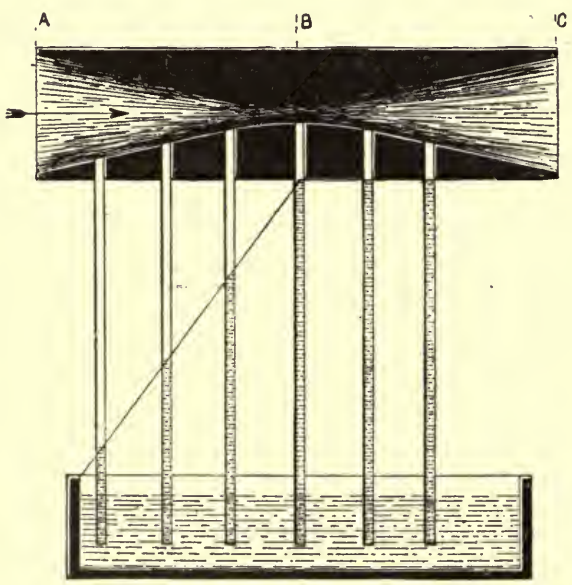


FIG. 111.—Experimental Nozzle.

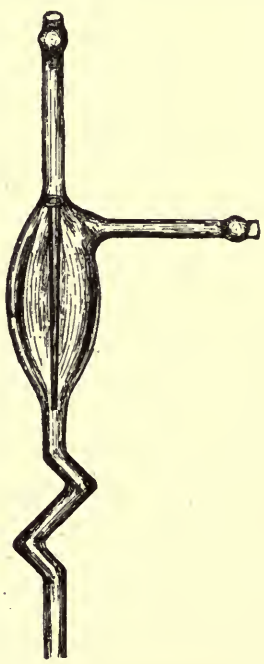


FIG. 112.—Filter Pump.

per second, we can also find that dimension. For the rest, we must depend upon experience. The fluid friction is reduced by tapering the nozzles; hence the tapers are of interest.

Consider Fig. 110, say, 1 square inch wide at the entrance A, and $\frac{1}{10}$ square inch at the throat C. Fluids liquid like water have considerable inertia, hence cannot without loss be started into quick motion in short time. The velocity of the water in the nozzle at C would be, we know, $= 8\sqrt{h}$, where h is the head corresponding to the pressure at C. Suppose the velocity at C = 100 feet per second, the section of C is equal to $\frac{1}{10}$ the section at A; hence the velocity at A will be 10 feet per second, and from that to B the same. At B the section begins to lessen and the velocity to increase, and the longer the cone from B to C the more gradual will the increase of velocity be. The velocity for water at A should not exceed 3 feet per second. The axial distance from B to C should not be less than twice the diameter at B for water.

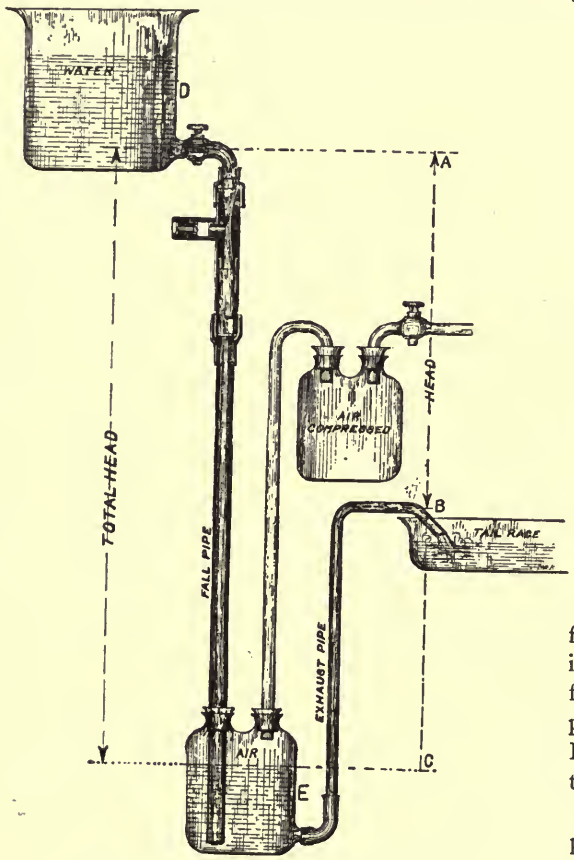


Fig. 113.—Exhauster and Compressor.

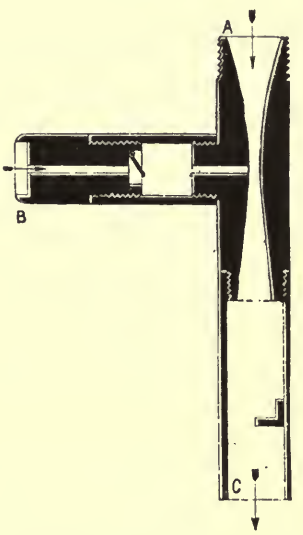


Fig. 114.—Filter Pump Nozzle.

A simple cone like this is employed only for throwing water, steam, or gases to a distance in a desired direction.

In Fig. 45 (p. 39) we showed that the pressure falls as the velocity increases, and *vice versâ* in a fluid flowing through the pipes at different sectional areas. This can also be shown for steam or air by an experiment as shown in Fig. 111, in which a bi-conical nozzle with small tubes at right angles to the axis is set up. The tubes dip into water or mercury some feet below the nozzle. On blowing air or steam through it is found that a partial vacuum is formed at the mouth of the tubes, and the amount of the vacuum may be measured by the height to which the liquid is sucked up into the tubes. The height will be greatest where the velocity of flow is greatest, that is at B and onwards. On this principle of suction many useful appliances are made, notably little exhausters for chemical work, one of which is shown in Fig. 112 for attaching to an ordinary water tap. The flowing water carries air with it and produces a vacuum in any vessel attached to the side pipe. It may also be used to produce air pressure, for

the water will carry the air into a receiver against considerable pressure. In order to do this the arrangement shown in Fig. 113 is made. The nozzle is prolonged and expanded trumpet shaped and fixed airtight into a large receiver with an overflow. The pressure at which the air will be delivered will depend upon the height to which this overflow is carried. Fig. 114 shows the nozzle separate on a large scale.

HYDRAULIC AIR COMPRESSOR

It is worthy of note that a large machine on this same principle has been designed and put into practice at a place called Magog in Canada. Instead of one side entrance for air, a great number of small pipes are let into the flowing water, just where it has acquired full velocity due to the height of the waterfall. It is the invention of Mr. Taylor of Montreal. The following is a description referring to the Fig. 115.

The letters on the figure are references to the parts as follows:—

A, penstock or water supply pipe.
B, receiving tank for water.
C, compressing pipe.
D, air chamber and separating tank.
E, shaft, or well, for return water. (The required pressure is proportional to the depth of the water in this shaft.)
F, tail race for discharge water.
G, timbering to support earth.
H, blow-off pipe.
I, compressed air main.
J, head piece, consisting of—
a, telescoping pipe, with
b, bell-mouth casting opening upwards.
c, cylindrical and conoidal casting.
d, vertical air supply pipes. (Each pipe has at its lower end a number of smaller air inlet pipes branching from it towards the centre of the compressing pipe.)
e, adjusting screws for varying the area of water inlet.
f, hand wheel and screw for raising the whole head piece.
K, disperser.
L, apron.
M, pipes to allow of the escape of air from beneath apron and disperser.
N, legs by which the separating tank is raised above the bottom of the shaft to allow of egress of water.
P, automatic regulating valve.

WORKING OF THE COMPRESSOR

The water is conveyed to the tank B through the penstock A, where it rises to the same level as the source of supply. In order to start the compressor the head piece J must be lowered by means of the hand wheel *f*, so that the water may be admitted between the two castings *b* and *c*. The supply of water to the compressor, and consequently the quantity of compressed air obtained, is governed by the depth to which the head piece is lowered into the water. The water enters the compressing pipe between the two castings *b* and *c*, passing among, and in the same direction as, the small air inlet pipes. A partial vacuum is created by the water at the ends of these small pipes, and hence atmospheric pressure drives the air into the water in innumerable small bubbles, which are carried by the water down the compressing pipe C. During their downward course with the water the bubbles are compressed, the final pressure being proportional to the column of return water sustained in the shaft E and tail race F.

The diagram Fig. 116 shows the relative sizes of the bubbles as they descend in a compressing pipe of 116 feet in length.

When they reach the disperser K their direction of motion is changed, along with that of the water, from the vertical to the horizontal. The disperser directs the mixed water and air towards the circumference of the separating tank D. Its direction is again changed towards the centre by the apron L. From thence the water flows outward, and, free of air, passes up the lower edge of the separating tank. During this process of travel in the separating tank, which is slow compared with the motion in the compressing pipe C, the air by its buoyancy has been rising through the water and pipes M, M, from under the apron and disperser, to the top of the air chamber D, where it displaces the water. The air in the chamber is kept under a nearly uniform pressure by the weight of the return water in the shaft and tail race.

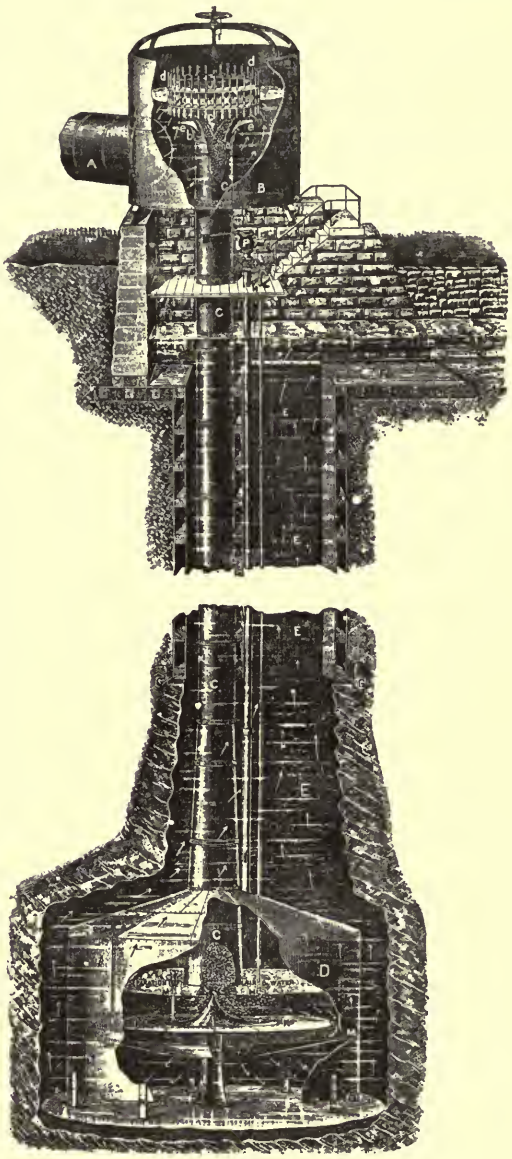


Fig. 115.—Hydraulic Air Compressor.

The air is conveyed through the main I, up the shaft to the automatic regulating valve (of which a diagram is given), and from thence to the engines, etc. The air pressure in the main and air chamber increases 1 lb. per square inch for each 2 feet $3\frac{1}{2}$ inches that the water is displaced downwards in the air chamber by the accumulating air. The variation in pressure from this source will not be more than 3 lbs. per square inch in a working plant. As the automatic valve requires a change of only 1 lb. per square inch pressure to close it completely, it will be evident that, by properly adjusting the valve, some air can always be retained in the air chamber, and that the water can be prevented from ever reaching the inlet to the air main. If a large quantity of air has accumulated in the chamber the valve allows of its free passage along the main; but when the air is being used more quickly than it is accumulating, and the pressure decreases below a certain point because the chamber is nearly emptied of air, the valve shuts partially, or completely, adjusting itself to the supply from the compressor.

When the air has displaced the water almost to the lower end of the compressing pipe it escapes through the blow-off pipe H.

In the hydraulic air compressor a bubble of air, while passing down the compressing pipe, is kept cool by the body of water surrounding it. The process of compression

is comparatively slow, occupying from 15 to 25 seconds. The temperature of the water is scarcely affected by the heat which it receives from the air during compression (the heat required to raise 1 cubic foot of water from one temperature to any other being about 3500 times as much as that necessary to produce the same change of temperature with 1 cubic foot of air). The bubble of air is compressed at a constant temperature (*i.e.* isothermally), the temperature being that of the water, and the excess of moisture, caused by the gradually increasing pressure, is deposited on the walls of the bubble. Thus it is evident that the air is collected in the separating tank at the low temperature of the water, and as dry as it is possible to obtain it. By a test made by Professor C. H. M'Leod, Ma. E., of M'Gill University, on 50 cubic feet of air hydraulically compressed to 52 lbs. gauge pressure, it was found that the expanded air contained only $\frac{1}{5}$ of the vapour usually contained in the atmosphere during fine weather (or 14 per cent. of saturation). This test was made while the compressor was delivering through the main 1500 cubic feet of air per minute.

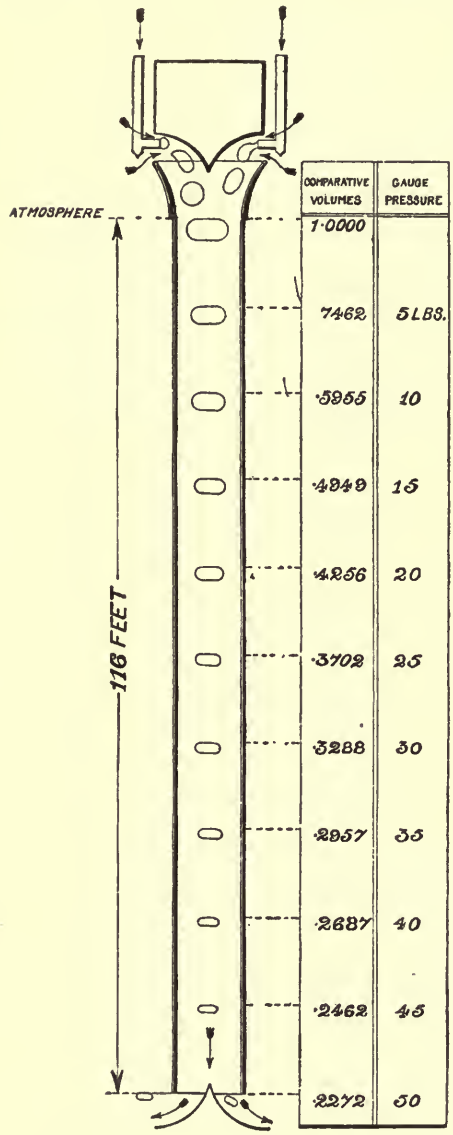


FIG. 116.—Compression of Air Bubble in Compression Pipe.

On 20th and 21st March 1897 a pair of 7 inch × 10 inch engines were run, under a full load, by the air delivered direct from the compressor, without preheating, for 40 hours continuously, without showing any sign of a moisture deposit on the interior of the exhaust.

At the same time the considerable fall in temperature from the expanding air was such as to produce, by condensation of the atmospheric moisture, a heavy coating of ice on the *outside* of the exhaust pipe.

The principles are simple. Referring to Fig. 113, the water will flow from the height A to B with a velocity proportional to $8\sqrt{h}$ – friction; the air drawn in will pass down with the water so long as the velocity of the water is greater than that of a bubble of air liberated at the bottom of the pipe, with the column of water at rest. This velocity is small, being less than that due to a head of 4 feet of water, being about 8 feet per second; hence a low fall will give sufficient velocity to the water to enable it to carry down the entangled air.

The pressure on the air is that due to the head BC, Fig. 113, so that we obtain higher pressure by sinking the vessel E in order to increase height CB.

There is a limit to the depth E may be sunk for any given head BA, depending upon the work lost in friction, agitation of the water, and air. And air is absorbed by the water, so that a considerable quantity is carried up the exhaust pipe and lost. Nevertheless it is a simple and useful machine, and will find applications in lands where waterfalls abound.

In these applications the fluids act by suction. In another class there is this same suction principle, and also another principle at work, in which a moving jet of fluid moving through another body of fluid induces motion therein, and the moving jet carries an annular body of fluid with it, and gives it velocity. On this principle Sir W. Siemens made an exhaust air ejector worked by steam. In order to give the steam jet a maximum of contact with the air to be set in motion, he made the steam jet a thin tube of steam produced between two converging nozzles. The air flows into a central nozzle and also into an outer nozzle, as shown in Fig. 117, so that there is a tube of air outside of the steam tube and a rod of air inside the steam tube. The combined air and steam flow away through an expanding tube, and so come to rest outside. As an example of its ejecting powers, a steam jet having a total sectional area of $\frac{1}{10}$ of a square inch, pressure 45 lbs., in 3 minutes reduced the pressure of air in a vessel of 225 cubic feet capacity to 15 inches of mercury.

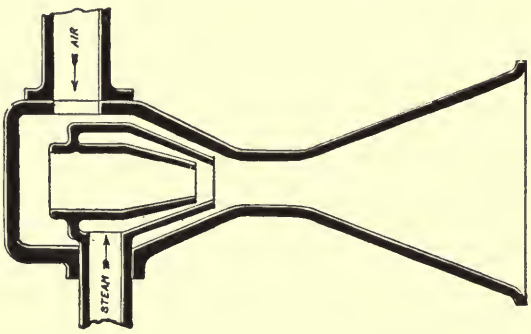


FIG. 117.—Siemens' Air Exhauster.

Jets can be arranged for exhausting or suction and for induction, or both. If we take two cones, as in Fig. 118, we shall find that there will be no suction produced by the steam jet, and it will not lift the water up into the water cone; for the orifice B is smaller than A. And hence as the velocity at A will be less than at B, and as the pressures are inversely as the velocities, there will be a pressure in A driving back the water.

Now, to get a greater velocity at A than that at B, we must expand the nozzle instead of contracting it as shown in Fig. 119. In this arrangement the steam from A

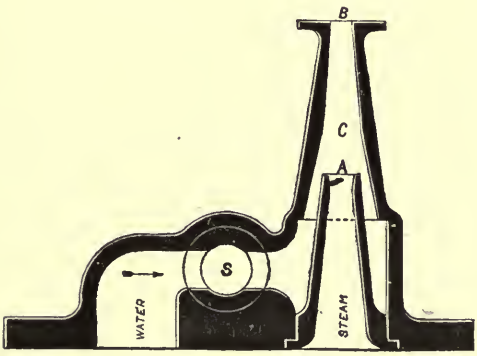


FIG. 118.—Non-lifting Jet.

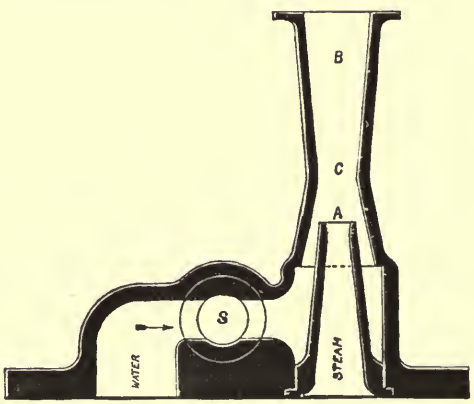


FIG. 119.—Lifting Jet.

will immediately induce the air to follow it out of the expanding cone, and a vacuum will rapidly be formed in the water pipe raising the water.

Hence we have two distinct combinations of cones. In Fig. 118 the water must be high enough to flow in itself, the jet will then force the water up to a high height or against a high pressure.

In the second combination the steam jet will induce a current of air at first, and, causing a partial vacuum, will lift the water and force it through B, but not to the same height or against the same pressure as Fig. 118; for the velocity of the water is reduced by widening the throat at C in order to get the highest velocity at A.

Whether the water enters under a head or is lifted up into a cone, the next effect we have to notice is, that the steam jet imparts an enormous velocity to the water, driving it from the second cone by pressure or impact, or both. There is no doubt about its being impulse from the steam jet upon the water which actuates the water jet, for steam at atmospheric pressure will deliver water against 70 lbs. above atmospheric pressure.

For instance, an exhaust steam injector, size No. 6, will deliver 100 lbs. of water per minute at 70 lbs. pressure, using in that time 8 lbs. of steam at atmospheric pressure. As 225 cubic feet of steam are required per minute to feed in 6000 lbs. of water per hour, a cubic foot at atmospheric pressure steam weighs .039 lb. The height of a water column of 70 lbs. pressure would be 163 feet, and the velocity of water $8\sqrt{160} = 8 \times 12.8$ = say, 100 feet per second. The velocity imparted to the water must be more than this by the amount required to overcome friction, etc.

The size of the delivery is about $\frac{1}{4}$ inch diameter, or .05 square inch; the weight of water to be delivered 100 lbs., or 1.6 lbs. per second. The kinetic energy would be $= \frac{V^2 w}{2g} = \frac{100^2 \times 1.6}{64.4} = 257$ foot-lbs. per second in the moving jet; but the whole work done is the forcing of the 1.6 lbs. against a head of pressure equal to the height of a water column exerting the pressure of the boiler, multiplied by 1.6, and added to the above. We can see that the first quantity is work done on the water to start it into 80 feet motion per second, from a state of rest; and the second quantity is work done to carry the water the height.

TABLE IX.

Size of Injector in Millimetres.	Pressure of Steam.						
	10 lbs.	30 lbs.	60 lbs.	80 lbs.	100 lbs.	120 lbs.	140 lbs.
	Delivery in Gallons per Hour.						
2	25	43	61	71	80	87	93
3	56	97	138	160	178	196	211
4	100	173	246	285	317	348	376
5	157	272	385	445	496	545	587
6	226	392	555	640	715	783	846
7	307	533	755	871	973	1067	1152
8	402	696	985	1137	1272	1393	1505
9	508	882	1247	1440	1610	1763	1905
10	628	1088	1540	1777	1987	2177	2352
11	760	1317	1863	2150	2405	2633	2846
12	905	1567	2217	2560	2861	3136	3387
13	1061	1840	2602	3005	3358	3680	3975
14	1231	2133	3018	3485	3895	4267	4610
15	1413	2450	3465	4000	4471	4900	5292
16	1608	2787	3942	4551	5087	5575	6022
17	1817	3146	4450	5138	5743	6291	6798
18	2036	3527	4990	5760	6438	7055	7633
19	2268	3930	5560	6418	7175	7871	8492
20	2513	4355	6160	7110	7950	8710	9410

A live steam injector delivers about same quantity with same size injector—0.25 inch diameter delivery cone = 0.05 square inch the quantity of steam would be $Q = AD \times 370$ = lbs. per minute, wherein Q is the quantity per minute, A the area, and D weight of steam per cubic foot at 80 lbs. pressure, 370 a constant. D at 80 lbs. pressure = 0.18.

Hence $Q = 370 \times 0.05 \times 0.18 = 3.33$ lbs. per minute, or $\frac{3.33}{60} = .055$ lb. per second to

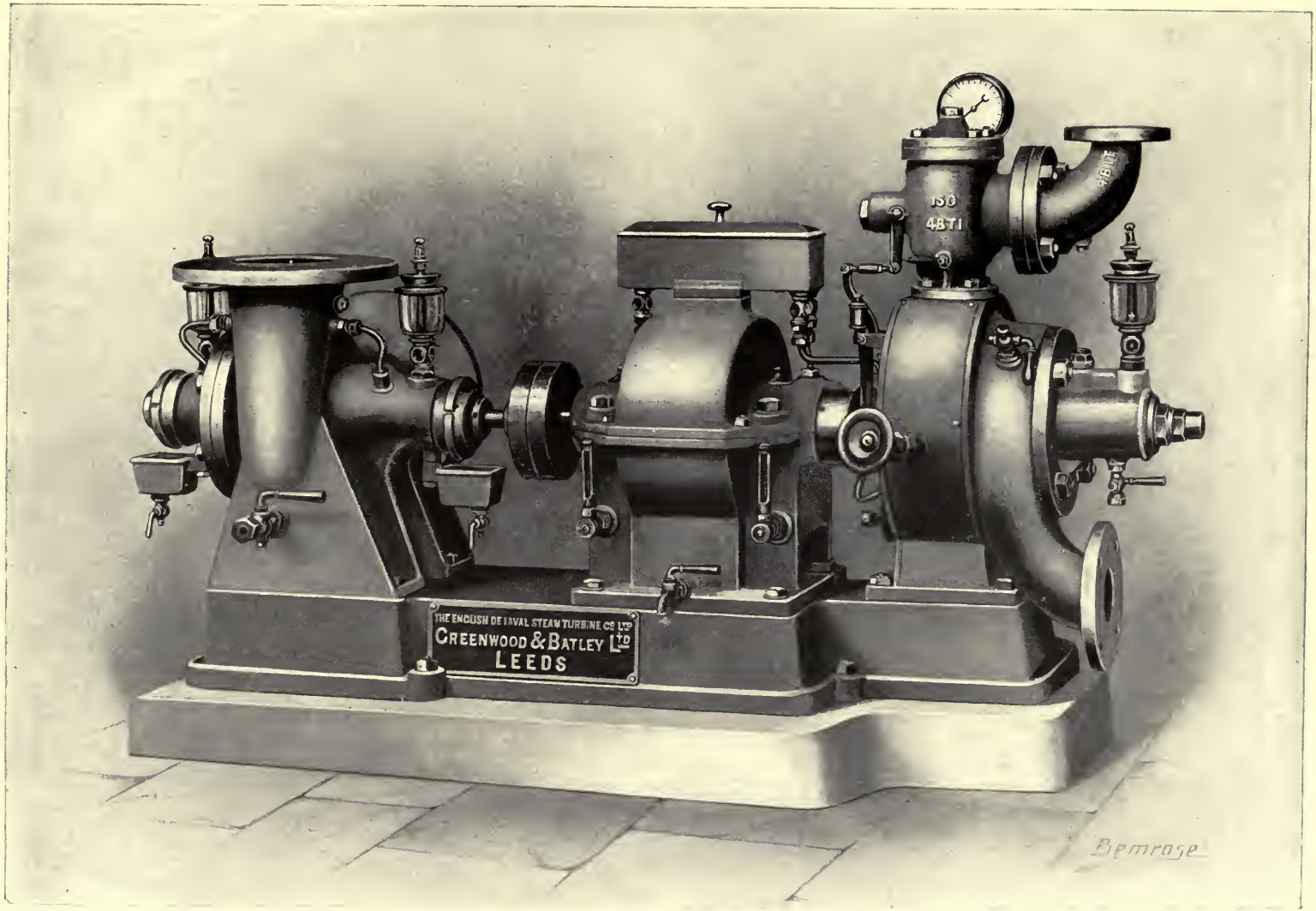


PLATE IV.—CENTRIFUGAL PUMP AND TURBINE COMBINED.

lift 1.7 lb. of water against 80 lbs. pressure. A better efficiency than for exhaust steam, from the point of view, as a water jet producer.

We have seen that the impulse of a steam jet is at the throat of a nozzle $\frac{1500^2 W}{2g}$; a jet like that would, if all its impulse were absorbed, give 20 horse-power.

3.33 lbs. of steam per minute would give 10 horse-power in a steam engine; in this injector it lifts 100 lbs. 185 feet in a minute, or 18,500 foot-lbs.—a little more than $\frac{1}{2}$ horse-power of work done. It is clear, then, that although "the impulse before impact must be the same as after impact" may be true, that very little of it is communicated to the water as impulse energy.

The fundamental equation is—

$$W = 1280 \times D^2 \sqrt{P}, \text{ or}$$
$$W = 1.98\, d^2 \sqrt{P},$$

where W = water weight to be injected per hour.
P = pressure in lbs.
D = diameter of throat in inches.
d = ,, ,, millimetres.

SIZES OF JETS

In approximate calculations for construction, if W = lbs. of feed water supplied per lb. of steam, then W + 1 will be the quantity entering the boiler.

The total heat of 1 lb. of steam reckoned from zero Fahr. $= h_1 x_1 L_1$.

Boiler pressure { wherein h_1 = sensible heat at the steam.
x_1 = dryness fraction.
L_1 = latent heat at the given pressure.

When the quantities are combined the total heat in the combining cone $= h_3$, the heat in the feed water $= h_5$.

The heat given up in condensation of the steam $= h_1 + xL_1 - h_3$ thermal units, and the heat gained by the feed water $= W(h_3 - h_5)$ units. W will therefore $= \frac{h + xL_1 - h_3}{h_3 - h_5}$.

For the velocity of the steam at the throat of the steam orifice we have an equation already, and may be taken as nearly constant at 1500 per second.

V_3, the velocity of the combined steam and water in the delivery

$$= \frac{V_2}{2(1+W)} + \sqrt{\frac{r_2^2}{4(1+W)^2}} \pm g\,H,$$

according as the feed water is above or below the delivery. If it is on a level with the delivery, $V_3 = \frac{V_2}{1+W}$.

If W_2 = the total water in lbs. per hour to be fed, the weight per second will be $\frac{W_2}{3600\,W}$, and the volume $\frac{W_2}{3600\,W} \times$ volume of 1 lb. of steam at the pressure given in the steam orifice $= V_2$; therefore $\frac{W_2 V_2}{3600\,W}$ = total volume.

The volume of steam passing the steam nozzle is AV_2 cubic feet, if A is given in square feet. V_2 we have taken as 1500. $A_2V_2 = \frac{W_2V_2}{3600\,W}$. Taking the value of V_2 as assumed, and πr_2^2 for A_2, we find that r_2 = the steam nozzle $= 0.113\sqrt{\frac{W_2V_2}{WV_2}}$. The combining cone orifice $= r_3 = 0.113\sqrt{\frac{1 - WW_2}{W_2PV_3}}$.

An example may be taken for 500 horse-power, boiler feed 10,000 lbs. of water

required per hour = W_2; feed water at 60° Fahr. feed water on the level, temperature of feed 170°, absolute steam pressure 100 lbs.

The quantity of feed per lb. of steam calculates out at 8.82 lbs. (in practice it is nearer 7). W = by equation = $\frac{298.09^\circ + 883 - 138.35^\circ}{138.35^\circ - 28^\circ} = 8.82$. Velocity of delivery is found by $V_3 = \frac{1500}{9.82} = 152$. Radius of steam orifice, $r_1 = 0.113\sqrt{\frac{10{,}000 \times 7}{8.82 \times 1500}} = 0.26 = .52$ inch diameter = 0.235 square inch.

For the radius of the combining cone orifice, $r_2 = 0.113\sqrt{\frac{9.82 \times 10{,}000}{8.82 \times 62.5 \times 150}} = 0.122$, or 0.244 inch diameter.

These are the fundamental dimensions from which the design can be completed.

The exhaust steam injector acts upon the same principles. The maximum flow of steam, we have already seen, occurs when $p - p_1$ is less than $\frac{3}{5}$ of p boiler pressure and p_1 the pressure blown into. Now, if we take exhaust steam at 15 lbs. above zero, and assume that the pressure falls to 8 lbs. in the combining cone, we would obtain the maximum weight flow of steam, for $\frac{8}{15}$ is less than $\frac{3}{5}$.

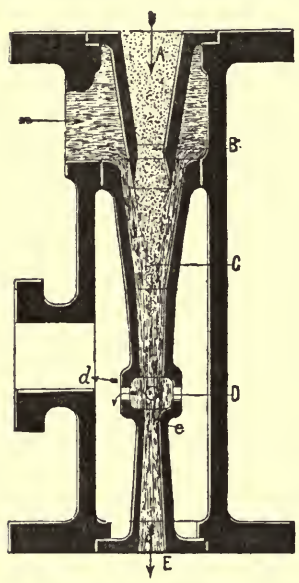


FIG. 120.—Loco Injector.

The volume of the steam per cubic foot is large, and so we must make the steam cone A larger in proportion, for it still takes 1 lb. of steam to raise up 7 lbs. of water. 0.038 is the weight of 1 cubic foot at atmospheric pressure; at 100 lbs. pressure in the example quoted the weight is 0.23, so that to admit the same quantity of steam the orifice areas A should be as the weights .23 is to 0.038, about 6 to 1 = 6 × .235 = 1.41 square inch. The water must have a slight fall into the injector.

Practically 1 lb. of steam raises 7 lbs. of water, and the steam cone would be 7 to 1 against a steam cone at 100 lbs. pressure; hence weight of steam Q = AD 370 = 1.635 square inch × 0.038 × 370 = 22.25 lbs. of steam per minute delivered, or 1335 lbs. per hour, and 1335 × 6 = 9345—very little less than the 10,000 delivered by the live steam. Where there are non-condensing engines exhaust steam can be used with advantage to raise large volumes of water.

The sensible heat of the exhaust steam plus the latent heat as it enters the steam cone is 1147, and that of the steam at 100 lbs. = 1171 per lb., so that if we take 1.022 lbs. of exhaust steam it will deliver the same heat in the combining cone as 1 lb. at 100 lbs. pressure, so that the fact that the exhaust steam will feed against 80 lbs. pressure, lifting 7 times its own weight, is not so mysterious.

We come now to examine a few typical injectors. For locomotive feeding they are incomparable. There is no lift in this case, and steam pressure need not vary much, so that a simple injector meets with great favour. Fig. 120 shows it in its simplest form. A is the steam cone, B the steam orifice, C the combining cone, in which the steam is seen as a cone wedging itself into the water, condensing and flowing across an air space D into the delivery cone E.

The air space D is of importance, as it provides a relief from back pressure which would arise if from any cause the quantity flowing at *d* could not get freely away though *e*. The flow of fluid must be unchecked, so as to maintain the velocity. Hence if a surplus of fluid were to flow and no opening D provided, the velocity would be checked. By providing an overflow any surplus is shaved off and escapes, so that no obstacle is presented to the full velocity from the combining cone.

In discussing cones and water lifting, we have shown in Figs. 118 and 119 that the combining cone outlet must be larger in area than the steam cone orifice, in order to start the injector on a lift of water. Sellars was the first to patent a device for this purpose, shown in diagram Fig. 121. He fits for this purpose a steam tight rod with a small nozzle drilled in its end marked the lifter. This nozzle slides by a screw in the steam cone, and gets its steam from holes drilled obliquely into the nozzle from the side of the rod in the position shown. It closes the main steam cone and delivers a small jet to lift the water and start with. When started the little nozzle is withdrawn by the screw, and its steam parts closed.

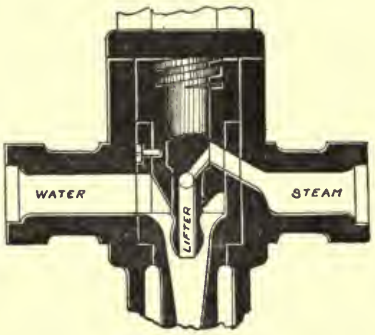


FIG. 121.—Lifting Nozzles.

A better method was introduced by Davis and Metcalfe in the flap nozzle injector (Fig. 122), wherein the nozzle is split and hinged so that when steam is turned on it opens up and presents an orifice greater than the steam cone, and so forms the vacuum to lift the water for starting. When the water arrives it is driven down in a contracting stream, and the flap closes upon it as a partial vacuum is formed around the stream of water. It is thus automatic, and if from any cause the injector should stop it will restart again. This instrument is well adapted for locomotives, as it requires no attention to keep it at work.

This restarting arrangement is also employed in the same makers' exhaust steam injector shown in Fig. 123, so that it starts off itself every time the engine is stopped and started. This injector is very simple and effective and easily adjusted. By turning the nut at the bottom the combining cone can be raised or lowered so as to nicely adjust

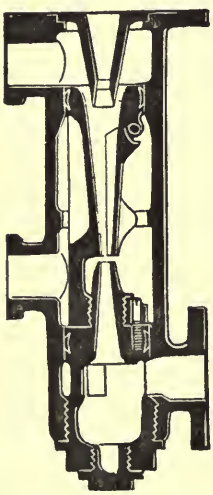


FIG. 122.—Self-starting Jet.

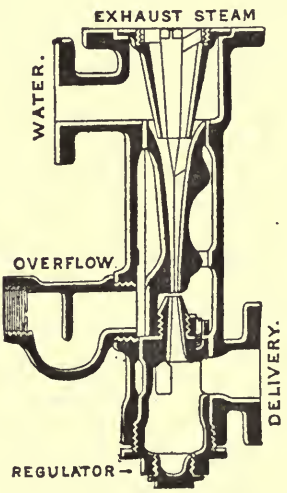


FIG. 123.—Exhaust Steam Injector.

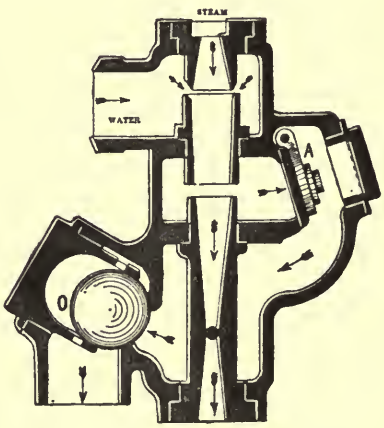


FIG. 124.—Automatic Restarting Injector.

the water supply. The conical rod in the steam cone causes the steam jet to enter as an annular jet, and so concentrates it upon the annular water jet. The steam in the centre of a solid steam jet not being so effective as that on its outer periphery, the division in the overflow is meant for a water seal to prevent inlet of air.

Another method for automatic restarting is shown in the latest Penberthy injector, Fig. 124. Although introduced by Mr. R. G. Brooke, we select the Penberthy as showing also the diverging steam cone, which must, as in the De Laval cone, increase

the impulse. The combining nozzle is nearly parallel, and cut into two lengths. In order to restart automatically, a flap valve A opens or closes a chamber around the division of the cone to the overflow O, and this valve opens and closes automatically. The ordinary overflow is also used at the narrow throat of the delivery cone.

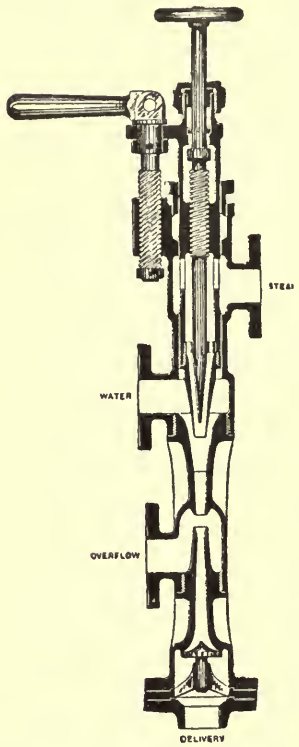


FIG. 125.—Giffard Injector.

There are many other designs for restarting automatically, all based on this principle of an automatic valve opening by the pressure to allow of the steam blowing out the air and the water to follow it. Immediately condensation occurs on arrival of the water, the valve closes by a partial vacuum forming.

The original injector, the Giffard, had a tapered rod to regulate the steam, and a moving cone to regulate the water. Wherever injectors have to work under different conditions of water and steam supply it is desirable to have regulation of this kind. The original Giffard injector is shown in Fig. 125. The steam cone slides by means of the side screw, and the tapered rod by means of the central screw and hand wheel. By moving either or both, adjustment can be made for a large variety of different conditions. In fact, it is an excellent machine with which to give or get an object lesson on the effects of varying pressures, temperatures, and lifts. It is, however, not restarting automatically.

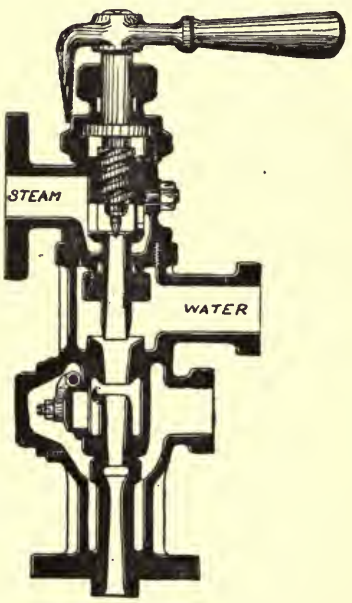


FIG. 126.—Adjustable Injector.

Mr. Brookes' automatic restarter, fitted with a sliding cone, which by moving the handle adjusts both steam and water simultaneously, is shown in Fig. 126. The combining cone is divided and fitted with a closing and opening automatic valve to the overflow. To make it restart, it has also the ordinary air space overflow. The steam cone is carried on a piston working in a cylinder, and can be screwed out or in by the quick thread screw. A conical end on the screw regulates the steam inlet, and the movement of the outlet orifice regulates the water, decreasing the one at the same time that the other increases. Hence a position can be found for satisfactory working over a long range of varying conditions.

COMPOUND INJECTORS

With a single injector the temperature of the feed cannot exceed 200° Fahr., because of the low pressures in the cones. Any higher temperature generates steam and destroys the action; but if, with one injector, we force feed water into another, we can, on account of the higher pressures, feed into the boiler at much higher temperatures.

We will illustrate one by Green & Boulding, of London, called the Buffalo

Injector, as a very well designed example. The first injector on the right-hand side is designed to lift the water and force it into the high pressure injector. On lifting the handle 13 (Fig. 128), spindle 2 lifts the steam valve through 21 and allows steam to flow through steam cones 24 and 22. The lifting injector draws water through 21 and forces it through combining and delivery cone 23, whence it enters combining cone 25, and from thence to the boiler. The overflow valve of the forcer is worked by spindle 17 by the side rod shown in elevation (Fig. 127), which allows it to close when the machine is at work, and opens it when stopped. When the water arrives and steam condenses a water pressure is produced in discharge of the forcer 25, which closes the overflow valve. This automatic closing of the valve is the feature of this injector.

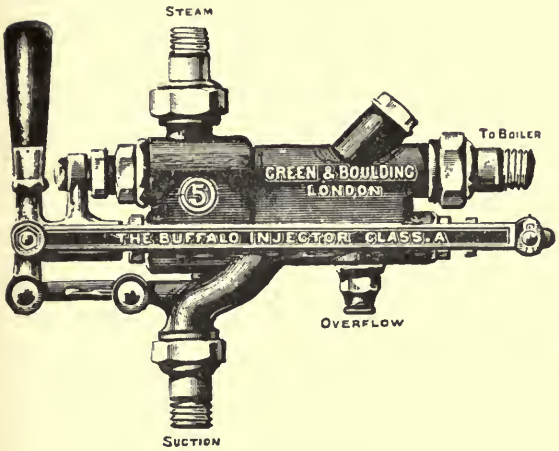


FIG. 127.—Compound Injector

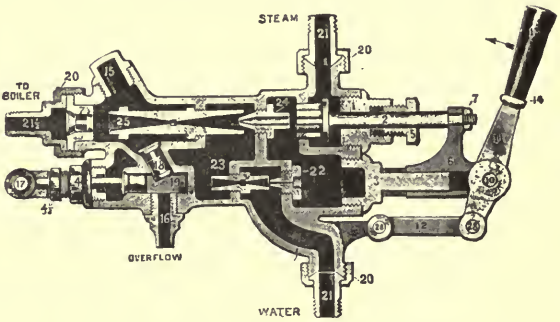


FIG. 128.—Section of Compound Injector.

Exhaust injectors can be combined with live steam injectors to feed into very high pressures. Messrs. Holden & Brookes' excellent arrangement is shown in Fig. 129. A jet of live steam starts the exhaust injector through pipe A, and it delivers at first partly through P and the live steam injector. When fully started, all the feed takes place through the live steam injector.

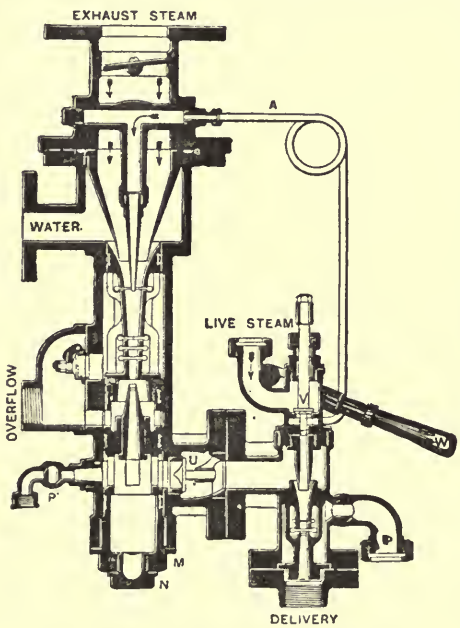


FIG. 129.—Compound Injector.

AIR EJECTORS

We have already referred to these. For some purposes they are admissible, such as on a locomotive, where a fan or pump would be objectionable on account of requiring attention. They are not economical. The only case of steam jets being employed with economy in prime movers is that of the De Laval turbine, where it is utilised by special means.

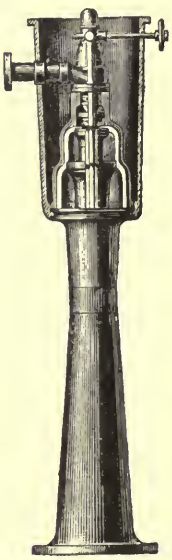


FIG. 130.—Air Ejector.

One application, however, deserves notice, that of furnace blowers where steam is plentiful and cheap, as it is where the waste heat generates it. Messrs. Korting Brothers supply a blower as shown in Fig. 130. On the bi-conical pipe is fixed a head

containing three or four cones, the one blowing into the other; and the top one a small steam cone. The object for using a number of cones in succession, each succeeding cone increasing in size, is to induce a large volume of air at low pressure. The steam sets up a flow in the first jet, this induces an additional flow in the next, and so on until the accumulated blast enters the large delivery pipe.

W lbs. of air are set in motion by 1 lb. of steam with a velocity V of steam and a velocity V_1 of the mixture, and the velocity of the air as it enters the cone by suction V_1, then the momentum is $M = \frac{V + WV_1}{1 + W}$. The head of steam is $\frac{V^2}{2g}$, and of the combined air and steam $\frac{V_1^2}{2g}$. The efficiency is difficult to determine accurately; theoretically, it should be $\frac{W\left(1 + \frac{V_1}{V}W_2\right)}{(1+W)^2}$. It therefore increases with the velocity of the air entering each jet. Hence increasing the velocity from one to the other in succession is a gain.

EJECTOR CONDENSER

These machines are designed to condense the exhaust steam from engines to form a working vacuum. They have the drawback of the jet condensers and others in mixing the condensed steam water with the condensing water. And therefore the pure condensed water is mixed with salt or impure water; whereas, in the surface condenser, canal, river, or sea water may be used for condensing, while the pure condensed water of the exhaust may be saved for boiler feeding. However, they have a large field of usefulness where these considerations do not apply with any force.

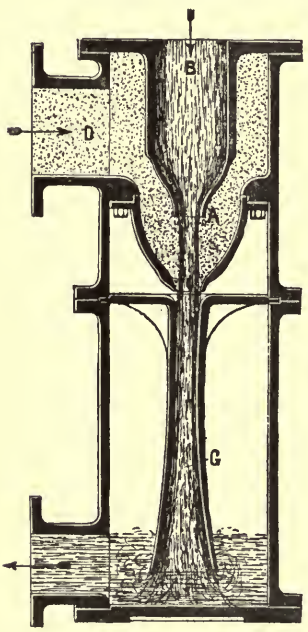


FIG. 131.—Morton's Simple Condenser.

This invention is due to Mr. Alexander Morton of Messrs. Morton & Thomson, Glasgow. In this condenser the proportion of water to steam required is much the same as in surface or jet condensers, about 27 lbs. of water to 1 of steam.

Morton, scientifically from the first, designed his ejectors to make use of the smallest head of condensing water,—in fact, it could be submerged in a river or pond of cold water and so draw in the condensing water. He designed them to lift the condensing water by means of a small supplementary live steam jet; also to regulate the water supply cones by movable cones; also a special automatic regulator to regulate the water supply. All these devices, however ingenious and in some few cases necessary, introduced the elements of complication, and attention was required for them.

In practice it is found better to provide an ejector condenser of sufficient water capacity for the maximum load, and a fair vacuum; and to maintain a head of at least 15 feet on the condensing water, if not by a natural fall, then by a centrifugal pump driven by power. For wherever there is an ejector condenser there will be power available.

With a fall of 16 feet the velocity of the water $= 8\sqrt{16} = 32$ feet per second. Hence it will balance the atmospheric pressure and produce a vacuum without the action of the exhaust, and maintain a good vacuum under all ordinary conditions. It is therefore usual now to employ a plain condenser, and in all cases provide a fall of condensing

water usually by pump. Morton's plain, simple fixed capacity condenser is shown in diagram Fig. 131. The water enters the cone B flowing out at A at about 32 feet per second, forming a rod of cold water. Rushing at this velocity, the exhaust enters at D, and surrounds this cold rod of water; it quickly condenses thereon, increasing its diameter slightly, and is carried with the rod right into G the delivery, from which it cannot return, for the pressure due to the velocity is greater than an atmosphere. Any air in the steam, if not of more than usual quantities in exhaust steam, is also carried down.

The Korting form of this ejector provides a longer contact for the steam and water, by using a long series of cones as shown in Fig. 132. I have used this condenser with great success in many cases, in nearly all of which the condensing water has been pumped up by centrifugals. For each lb. of steam we allow 27 lbs. of water lifted 16 feet, and allowing 25 lbs. of steam per horse-power. Thus we get the power required at the pump = 25 × 27 = condensing water per horse-power = 675 lbs.; and 675 × 16 = foot-lbs. = 10,800 per hour, or 180 foot-lbs. per minute per horse-power. As the friction and other losses amount to nearly as much, it is safe to calculate on 360 foot-lbs. per minute per horse-power, or roughly about 1 per cent. of the engine power.

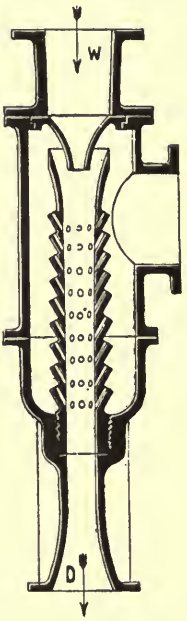


FIG. 132.—Korting Ejector Condenser.

A good vacuum increases the power of steam turbines very largely, so that it is good economy even to provide an artificial head and cooling apparatus for condensing water; in which case a simple ejector condenser is by far preferable to any other.

A water pressure intensifier by means of a water jet has been used to increase the head of water in town supply pipes, in case of a fire requiring a jet beyond the reach of the ordinary head. This practice is, however, only possible where a high pressure supply exists alongside of the ordinary supply. The jet is shown in Fig. 133.

In London, Glasgow, and other large towns a hydraulic pressure supply at 750 lbs. is laid on for working lifts and high pressure rams and presses. This supply can be led into an ordinary supply pipe A in a very fine jet, as shown in E, and the flow of water accelerated, and the pressure increased in the fire extinguishing jets. It requires care in using the jet. The delivery jets must all be freely opened before the high pressure jet is opened, and they must not be closed before the high pressure jet is shut off.

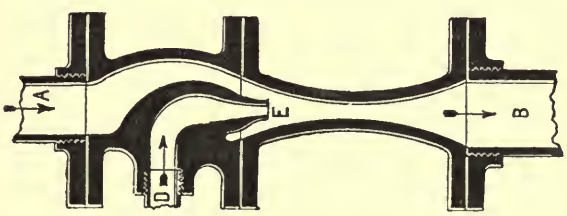


FIG. 133.—Water Pressure Intensifier Jet.

In Fig. 134 we give a section of the Penberthy No. 2 water jet lifter, showing the expanding steam nozzle. They are convenient in some cases for emergency and temporary use; but all water lifters by steam jets are wasteful of steam, and therefore are not adopted for regular pumping of water to higher levels.

We shall conclude our examination of fluid-upon-fluid engines and machines by a brief reference to ship propulsion by fluid or jets, a method which is worthy of consideration along with the only alternative, the screw propeller; which, in fact, is only a pump outside the vessel producing a water stream, the reaction of which propels the boat.

The propulsion of steam vessels by water jets occupied the attention of the British Government's Admiralty some thirty years ago. They were tried on a vessel called the

Waterwitch with some success, but the experiments for some reason or other were not concluded, and no report ever issued of results.

Mr. J. R. Ruthven gave a brief report, which was read before the Society of Engineers and Shipbuilders, Glasgow, 27th October 1891, from which we gather :—

The experimental trials made were those of the *Waterwitch*, a gunboat built for the Admiralty, with the purpose of comparing the propelling and manœuvring power of the jet with the screw. For this purpose two screw ships, the *Viper* and *Vixen*, twin screw vessels of about the same tonnage and same midship section, were built as fair competitors with the jet propelled *Waterwitch*.

The water jets are obtained by a centrifugal pump which takes its supply from the sea, or, in case of heavy leakage, from the hold of the ship. The centrifugal wheel is placed horizontally in the ship, and is driven direct by steam engines working horizontally.

There is an arrangement to reverse the direction of the discharge of the jets, so that the vessel can be propelled ahead by discharging the water astern, or astern by discharging ahead, or the jet at one side of the vessel may discharge ahead, and on the other astern ; in this case the ship neither goes ahead nor astern, but turns round in her own length.

The twin screw vessels *Viper* and *Vixen* and the jet propelled *Waterwitch*, common ships, all of the same midship section, and nearly same displacement. The twin screw vessels had double stern posts and two rudders at the stern, the *Waterwitch* had two rudders, one at the bow and one at the stern, each working in a heavy frame, intended to be used as a ram, at either end. This must have caused considerable increased obstruction, and as the ship is longer, on account of these rudders and frames, more difficult to turn in her own length.

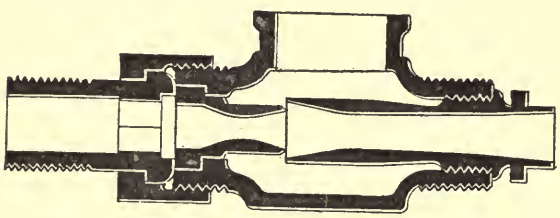


Fig. 134.—"Penberthy" Water Jet Lifter.

In 1863 it was decided, by the Admiralty, to test the water jet on a large scale.

Some time before this date the Admiralty had been satisfied on the feasibility of the plans, by themselves making and applying the jet, in an old screw gunboat, the *Jackdaw*, at Devonport, by means of a series of teeth wheels and shafts, working from the old screw shaft to the pump, which had only one outlet led to one side of the ship. With this imperfect arrangement the ship was propelled, and a speed was attained which satisfied the Admiralty that it gave promise of a success.

An important point in connection with the jet as a propeller must be noted, that is, its power in stopping the vessel. This was tried with the *Waterwitch*, and from going full speed ahead the order was given to reverse the action of the jets, and the vessel was stopped in nearly her own length. This was ascertained by throwing a billet of wood square from the bow on the order given to reverse the jets, and the ship was stopped when the floating billet was level with the stern. This is a very remarkable property, which the screw cannot approach.

This power of quickly stopping and reversing a ship's headway must be of great importance to the safety of the ship, by enabling her in many cases to avoid collisions which the continual increase of speeds makes more and more dangerous. In passing, it may be remarked that the great stopping power is a proof of the efficiency of the water jet as a propeller, the value of which is obvious in quickly stopping and backing, while with the great power of turning there is no danger from loss of steerage way, as the ship can be effectually steered by the jets when she has little or no way on her, at which time, of course, the rudder is of little or no use.

TABLE X.

Vessel's name	*Vixen*		*Viper*				*Waterwitch*							
Nominal horse-power	160		160				160							
Date of trial	2nd Aug. 1867		5th Aug. 1867		11th Oct. 1867		9-10th Aug. '67		28th Aug. 1867		3rd Sept. 1867		12th Oct. 1867	
Draught { Fore	Ft. In. 9 10		Ft. In. 9 11		Ft. In. 9 11		Ft. In. 10 9		Ft. In. 10 7½		Ft. In. 9 9		Ft. In. 10 7	
Draught { Aft	11 11		11 10		11 10		11 8		11 1½		11 9		11 2	
Midship Section, square feet	336		335·7		336		347		336		333		336	
Boiler power	Full	Half	Full	Half	Full	Half	Full	Half	Full	Half	Full	Half	Full	Half
Revolutions	108.6	85.1	109.2	82	110.5	83.7	40.7	26.2	41.5	28.2	40.9	27.9	40.4	30.6
Indicated horse-power	657	326	652	334	696	344	777	226	801	271	769	268	759	348.5
Speed, knots	9.06	7.347	9.475	7.334	9.586	7.625	9.237	6.227	9.219	6.047	8.88	6.326	9.299	7.206
$\frac{\text{Speed}^3 \times \text{Midship Section}}{\text{I.H.P.}}$	379.9	395	438.5	396.5	435.4	431.3	351.9	369.6	328.7	273.9	303.3	314.3	357.8	359.6

In the discussion upon this paper it is interesting to hear what her commander said about her.

Captain Sharpe, R.N., said: "I can safely say, having commanded the *Waterwitch* during the time the experiments took place, I always urged (and still think) that they were never carried on to the extent that they might have been. Whilst in command of the *Waterwitch* I felt the extreme advantage of having her always under my own individual command. I could stop her, go astern, turn the ship, or do anything without reference to the engineers. The only thing I have to add is that during the time I was in the ship I was very sorry to think that so little use came of it, and that so little has since been done."

The subject has dropped, although there are some small steamers of the lifeboat class still made with jet propellers, and the introduction of high speed turbines has proven that there are difficulties with propellers of the screw type; hence jets may after all be reverted to. It will therefore be of interest to investigate the matter a little further, with thirty years' more experience and progress than Mr. Ruthven had to aid him.

The late Mr. Miller, secretary of the Society of Engineers and Shipbuilders, Glasgow, also read a paper on jet propellers. His proposal was not made as a serious attempt at ship propulsion in general, but with a view to assist in the case of a failure of the screw, its shaft, or engines. So long as the boilers were intact he calculated that a steam jet, delivering astern all the steam the boilers could make, would propel her by reaction at 3 knots an hour, a speed not much but better than nothing at all. The difficulty, however, arises, that if far from land she would burn up all the fuel long before getting to port, and therefore never could finish the voyage. If she had coal for 12 days at 12 knots steaming, she could not travel far at 3 knots on the coal in stock.

The screw propeller is a pump outside of the vessel, which thrusts the water away, and the reaction propels the ship. The jet propeller driven by a pump does the same thing exactly. The only difference is, the pump is inside in one case, and outside in the other.

With the outside pump there are many difficulties, well known. The loss of the propeller is not uncommon, neither is the shaft breaking uncommon. And as the screw is necessarily a slow speed pump, the shaft has become of enormous weight and diameter. With the inside pump and propeller jets high speed pumps can be used. There are no propellers or shafts liable to be lost or broken. With obvious advantages it, however, has some natural limits which may keep it from extensive adoption.

It is all a question of velocities and pressures to be properly apportioned. For instance, in Mr. Miller's proposal to use a large steam jet, to get any efficiency, the ship

should move at some large fraction of velocity of the jet at the orifice. This velocity is about 1500 feet per second, or about 17 miles a minute.

The water jet, then, as a very first requisite, must have a velocity about equal to that of speed desired, and this limits its efficiency and usefulness.

Suppose the speed to be fixed at 10 knots per hour, it has been found that at 5 knots and upwards the power required is as the square of the speed nearly; for our present purposes that is at any rate assumed. At 5 knots it takes $\frac{1}{6}$ of a lb. pull or push to propel a vessel for every square foot of wetted surface, so that if we assume a ship with 6000 square feet of wetted surface it will require 1000 lbs. thrust to keep up 5 knots. Then if 10 knots are required, we get $\frac{10^2 \times 1000}{5^2} = \frac{100,000}{25} = 4000$ lbs. thrust for 10 knots.

10 knots is a rate of travel = 60,870 feet per hour, and $\frac{60870}{3600}$ = nearly 16 feet per second. Now, this is the point where a limit comes in. This speed, 16 feet per second, is the speed of efflux which the jet should have to get 100 per cent. efficiency. We can calculate the head or pressure of water which will give this velocity. $H = \frac{V^2}{2g} = 4$ feet head, or about 2 lbs. pressure. Now, if we divide the total thrust required by this pressure, $\frac{4000}{2} = 2000$ = the area of jet in section in square inches, $\frac{2000}{144} = 13.2$ square feet. A pipe of about 50 inches diameter would give this area.

Hence to get efficiency we require a large pump, and so large that we put it outside in the form of a screw in screw propellers.

But the pressure is small, 2 lbs. or 4 feet head, so that although the velocity of the water jet issuing from the ship is limited to this velocity, we might run a small centrifugal pump at a high velocity, and the water may have a greater velocity in the pipes inside the ship. 13 square feet of area × 16 feet velocity = 13 × 16 = 208 cubic feet of water per second, and 208 × 62.5 = 13,000 lbs. of water per second, 52,000 foot-lbs. per second. Nearly 100 horse-power would be required in the water, and probably the efficiency would be less than 50 per cent., so that an indicated horse-power would be required of more than 200 in all.

Now, suppose we double the pressure to an 8-foot head, then velocity $8\sqrt{8} = 2.83 \times 8 = 22.64$ feet per second velocity. With double the pressure we do with half the jet area,—6.5 square feet, or 1000 square inches. A 32-inch pipe or jet would now suffice.

The velocity is 22 feet, and this multiplied by 6.5 = cubic feet of water per second = 143 cubic feet, 9000 lbs. nearly. And energy per second = 9000 × 8 = 72,000 foot-lbs. = 130 horse-power in the water, so that the speed of the vessel would be still 10 knots; but instead of 100 horse-power, it would now require 130. The higher the velocity of the jet the less the efficiency, unless the speed of the ship is increased at same rate.

We thus find that to give the jet propeller a chance we require to move a very large volume of water at a speed above that of the vessel, and the centrifugal pump seems to be the best machine for moving these large quantities at low pressure. Now, the steam turbine coupled to a modern centrifugal pump would be something very different to the engines and pump used in the early tests just referred to. 80 per cent. efficiency can be realised, that is, 100 horse-power turbine pump will give 80 in the water horse-power. We cannot in this place enter into the whole question of centrifugal turbines and ship resistances; we are considering water jet fluid-on-fluid engines.

The next observation to be made on the subject is that the pump ought not to throw the water out through the jets directly. The object of the pump is to produce the desired pressure on the water, and for this purpose the water when it leaves the pump

should be delivered to a chamber or tank through widely diverging nozzles, so that it is without agitation brought to rest or to a very slow motion.

The propulsion is due to the reaction of the jets, to the unbalanced pressure of the water in this tank. If we have a tank of water maintained at 5 or 6 lbs. pressure, and open a hole in one side 2000 square inches in area, we get an unbalanced pressure of 6 x 2000 of 12,000 lbs. on the opposite side of the tank to the opening, and that is the force which propels the ship.

By careful design an efficiency as high as that of the screw propeller may be expected. At any rate, it is worth investigating again practically, whether with modern steam turbines driving modern centrifugal or screw pumps, and with fluid-flow and pressures directed and adjusted on modern scientific principles, that the jet is not a better instrument than the screw for propulsion. The steam turbine has so much to recommend it for ship propulsion; yet its full advantage can never be realised on small high speed screws, and to gear down to large low speed screws would certainly be a step backwards.

High speed is the order of the day, and the higher the speed the better for jet propelled vessels, whereas at high speeds the screw falls off in efficiency. The weight and space occupied would be much less, and the advantages of having no outside propeller liable to be lost or damaged, and no tunnel shafting, are not to be despised.

The problem would be still further simplified if an efficient steam jet could be used to propel these large volumes of water. The pumps and turbines would then be abolished also. But water jets propelled by steam jets are very inefficient, pretty much for the same reasons that the jet propeller failed of old. The steam has a speed of, say, 1500 in an injector cone, and the water to have any efficiency would require to move at a large fraction of this speed to get economy; but no water jet can be induced to move at such velocities. At 170 lbs. pressure the water would issue from the delivery cone at $8\sqrt{230}$ = 120 feet per second, so that if the steam jet in the injector flowing out at 1500 or more per second, succeeds in only imparting 230 feet per second to the jet from the combining cone, it will feed the boiler; but 230 is a small fraction of 1500. For boiler feeding this inefficiency matters not. The whole steam, with all its heat, minus that radiated and conducted, and other losses, re-enters the boiler, so that heat efficiency does not matter.

Theoretically, the velocities imparted should be inversely as the weights in motion. If 1 lb. of steam flowing at 1200 feet per second struck 11 lbs. of water, the result should be 12 lbs. of water moving at 100 feet per second, and so the kinetic energy of the 12 lbs. of water at 100 feet would be 1200, same as 1 lb. of steam at 1200 feet; for the total momentum before impact must equal the total momentum after impact.

If this could be realised in practice, then it would be useless to go on making reciprocating rotary or turbine steam engines. A Pelton wheel driven by this ideal water jet would far and away excel the lot. But the steam condenses instantly, and so softens the blow of impact, and the water has considerable inertia, so that it lags behind the rushing jet. Consequently, a steam jet as a water lifter is a poor affair from an economical point of view.

Recently Penberthy made some improvement in the direction of applying the De Laval cone to injectors. In all early injectors and water jet lifters the steam cone converges to the very end; but some makers commenced to diverge the cone a little, as shown in Fig. 126, evidently feeling their way cautiously. Penberthy has carried this diverging of the steam cone still further, as shown in Fig. 124, in his latest injectors and water lifters, with considerable improvement; how much we cannot say, for it is very difficult to get reliable information about jet pumps and injectors.

However, I made some experiments myself, using properly designed diverging steam cones, and thereby more than doubled the water lifted by the same steam consumpt. The reason is that the steam has a velocity nearly double of that issuing

from a converging nozzle, and the heat has been utilised more efficiently before condensation.

We have seen that compound injectors have been used to feed high pressure boilers at high temperatures.

A compound water lifter with several nozzles in series has also been devised whereby enormous velocities could be obtained. The water is started from rest in the first

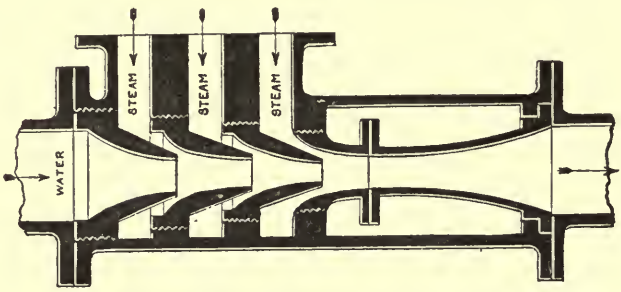


FIG. 135.—Multiple Nozzle Water Lifter.

cone, accelerated in the second, third, and so on, until it acquires a velocity approaching that of the steam jet in the last one; it is hoped that an efficient water lifter will result, water being accelerated in stages beginning with a small jet of steam. That is, instead of all the steam passing in one annular jet of, say, $\frac{1}{10}$-inch sectional area, it is passed in five each $\frac{1}{50}$ inch area, or three of $\frac{1}{30}$ inch each. The cones are reduced in proportion as the velocity is increased; it is shown in section, Fig. 135.

In recent tests the actual results of a turbine centrifugal pump combination for water pressure, the following table of results was obtained. It shows what might be done now with jet propellers with efficient pumps.

TABLE XI.—RESULTS OF TESTS WITH DE LAVAL STEAM TURBINE PUMPS.

Type of Turbine Pump.	Revolutions per Minute.	Height of Suction in Feet.	Height of Delivery in Feet.	Quantity of Water delivered per Second-gallons.	Water Horse-power.	Brake Horse-power.	Efficiency.
50 horse-power duplex pump coupled in parallel . .	1500	16.4	16.4	63.5	37.87	50.3	0.753

635 lbs. of water raised 32.8 feet = 37 horse-power in the water.

The pressure of the jet corresponding to the height would be 15 lbs., the velocity $= 8\sqrt{32.8} = 5.74 \times 8 = 45.76$ feet. The quantity of water is about 10 cubic feet, velocity 45 feet; hence $\frac{10}{45} = 0.22$ square feet, section of jet 31.68 square inches.

It is intended only in this brief discussion to put the jet propeller in its true position. Because some engineers are quite satisfied with screw propellers and unreasoningly refuse to consider anything else, that is no reason why the case for other propellers should not be put fairly before those of a more inquiring and enterprising disposition. Figures can only demonstrate the feasibility of the jet system, practice alone can decide for or against it; and what practical tests have been made were made with machinery —certainly not the best at the time, and far inferior to what would be employed to-day to make practical trials. The subject will be more fully discussed under "Marine Engines."

TABLE XII.—HORSE-POWER OF 1 CUBIC FOOT OF WATER PER MINUTE UNDER HEADS FROM 1 TO 1100 FEET.

Head in Feet.	Horse-power.	Head in Feet.	Horse-power.	Head in Feet.	Horse-power.	Head in Feet.	Horse-power.
1	.0016098	170	.273666	330	.531234	480	.772704
20	.032196	180	.289764	340	.547332	490	.788802
30	.048294	190	.305862	350	.563430	500	.804900
40	.064392	200	.321960	360	.579528	520	.837096
50	.080490	210	.338058	370	.595626	540	.869292
60	.096588	220	.354156	380	.611724	560	.901488
70	.112686	230	.370254	390	.627822	580	.933684
80	.128784	240	.386352	400	.643920	600	.965880
90	.144892	250	.402450	410	.660018	650	1.046370
100	.160980	260	.418548	420	.676116	700	1.126860
110	.177078	270	.434646	430	.692214	750	1.207350
120	.193176	280	.450744	440	.708312	800	1.287840
130	.209274	290	.466842	450	.724410	900	1.448820
140	.225372	300	.482940	460	.740508	1000	1.609800
150	.214170	310	.499038	470	.756606	1100	1.770780
160	.257568	320	.515136				

TABLE XIII.—SHOWING HEAD AND PRESSURES OF WATER.

Head of Water in Feet.	Head of Water in Yards.	Head of Water in Fathoms.	Head of Water in Metres.	Pressure of Water in Lbs. per Square Inch.	Head of Water in Feet.	Head of Water in Yards.	Head of Water in Fathoms.	Head of Water in Metres.	Pressure of Water in Lbs. per Square Inch.
5	1.66	0.83	1.52	2.16	185	61.6	30.8	56.3	80.1
10	3.33	1.66	3.04	4.33	190	63.3	31·6	57.9	82.3
15	5.00	2.50	4.57	6.49	195	65.0	32.5	59.4	84.4
20	6.66	3.33	6.09	8.66	200	66.6	33.3	60.9	86.6
25	8.33	4.16	7.62	10.80	205	68.3	34.1	62.4	88.8
30	10.00	5.00	9.14	12.90	210	70.0	35.0	64.0	90.9
35	11.60	5.83	10.60	15.10	215	71.6	35.8	65.5	93.1
40	13.3	6.66	12.1	17.3	220	73.3	36.6	67.0	95.3
45	15.0	7.50	13.7	19.4	225	75.0	37.5	68.5	97.4
50	16.6	8.33	15.2	21.6	230	76.6	38.3	70.1	99.6
55	18.3	9.16	16.7	23.8	235	78.3	39.1	71.6	101.8
60	20.0	10.0	18.2	25.9	240	80.0	40.0	73.1	103.9
65	21.6	10.8	19.8	28.1	245	81.6	40.8	74.6	106.1
70	23.3	11.6	21.3	30.3	250	83.3	41.6	76.2	108.3
75	25.0	12.5	22.8	32.4	255	85.0	42.5	77.7	110.4
80	26.6	13.3	24.3	34.6	305	101.6	50.8	92.9	132.1
85	28.3	14.1	25.9	36.8	310	103.3	51.6	94.4	134.3
90	30.0	15.0	27.4	38.9	315	105.0	52.5	96.0	136.4
95	31.6	15.8	28.9	41.1	320	106.6	53.3	97.5	138.6
100	33.3	16.6	30.4	43.3	325	108.3	54.1	99.0	140.8
105	35.0	17.5	32.0	45.4	330	110.0	55.0	100.5	142.9
110	36.6	18.3	33.5	47.6	335	111.6	55.8	102.1	145.1
115	38.3	19.1	35.0	49.8	340	113.3	56.6	103.6	147.3
120	40.0	20.0	36.5	51.9	345	115.0	57.5	105.1	149.4
125	41.6	20.8	38.1	54.1	350	116.6	58.3	106.6	151.6
130	43.3	21.6	39.6	56.3	355	118.3	59.1	108.2	153.8
135	45.0	22.5	41.1	58.4	360	120.0	60.0	109.7	155.9
140	46.6	23.3	42.6	60.6	365	121.6	60.8	111.2	158.1
145	48.3	24.1	44.1	62.8	370	123.3	61.6	112.7	160.3
150	50.0	25.0	45.7	64.9	375	125.0	62.5	114.3	162.4
155	51.6	25.8	47.2	67.1	380	126.6	63.3	115.8	164.6
160	53.3	26.6	48.7	69.3	385	128.3	64.1	117.3	166.8
165	55.0	27.5	50.2	71.4	390	130.0	65.0	118.8	168.9
170	56.6	28.3	51.8	73.6	395	131.6	65.8	120.3	171.1
175	58.3	29.1	53.3	75.8	400	133.3	66.6	121.9	173.3
180	60.0	30.0	54.8	77.9					

THE CENTRIFUGAL PUMP

This useful apparatus might come under the head of pumps, but as its action depends on fluid velocities and pressures it may be as well to include it in this chapter. The principle of the machine has been given in its simplest form by Professor Goodere, as follows.

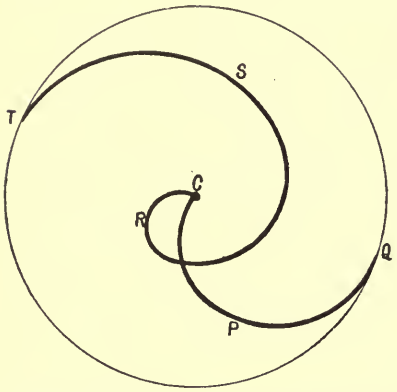


FIG. 136.—Theoretical Pump Experiment.

Conceive a ring threaded on a rod, pointing towards and at right angles from a shaft, and revolving about it. We know that the rod will continually press the ring in a line perpendicular to the rod. The ring will tend always to go forward in a straight line.

We have now to refer to a simple experiment in the geometry of motion. Let a circular disc (Fig. 136) be rotated about its centre C, mark that centre with the point of a pencil, and then draw the pencil rapidly from the centre outwards to the circumference. A curve will be traced on the disc, which will take the form CPQ if the disc rotates slowly, or will become more spiral in character, as shown by CRST, when the velocity of rotation is increased. The pencil moves radially in a straight line, but it traces a curve by reason of the increasing linear velocity of each point in the circle as we pass from the centre outwards. Our conclusion is, that a curved rod will act more effectively than an inclined rod, and that if the curvature be regulated to the velocity of rotation a sustained and uniform push may be maintained on each portion of water as it passes from the centre to the circumference.

In constructing a pump on this principle we begin with a circular disc AB, shown in section (Fig. 137), which receives the water pressure on both sides. This is an example of *balanced pressure*, and the friction is correspondingly reduced. The disc forms the central division of a hollow circular box, the water enters on both sides, as shown by the arrows, and is forced outwards by the curved vanes, the three successive types of vane are (1) radial, (2) inclined, (3) curved. In a pump by Mr. Appold the diameter of the case is 12 inches, that of the central opening is 6 inches, also there are 6 arms,

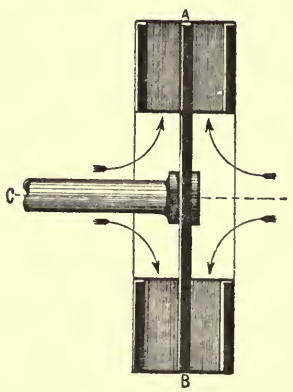


FIG. 137.—Centrifugal Pump.

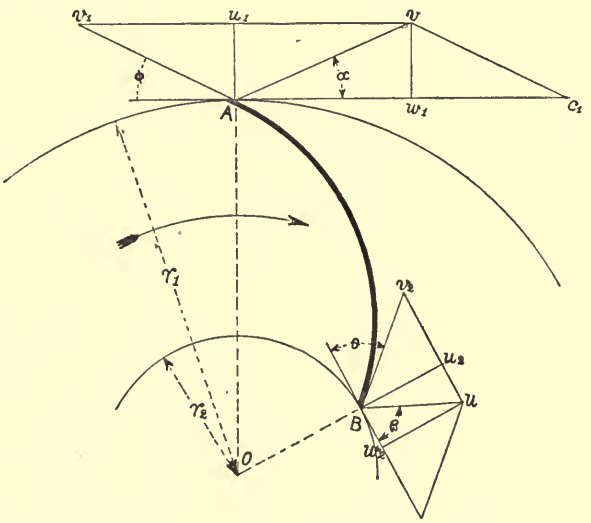


FIG. 138.—Construction of Centrifugal Pump Blades.

curved backwards and terminating nearly in a tangent to the circumference of the bounding circle.

The pump is set into rapid rotation between flat cheeks at the bottom of a pipe, the circumference of the box being open to the inside of the pipe, and the centre being open to the supply of water. The height to which they lift is equal to the square of the peripheral speeds in feet per second.

To lift the water to any given height theoretically it must be given a centrifugal force equal to $\sqrt{2gH}$ where H is the height, or, approximately, $8\sqrt{H}$. To obtain this force the speed of the periphery of the fan blades must be run at speeds depending upon their curvature and the angles which they make with the outer and inner peripheries. The diagram Fig. 138 shows the usual geometrical construction and parallelograms for the entering ends and leaving ends of the blades. Referring to the diagram Fig. 138, the speed of the blade at A depends upon the angle ϕ between the prolongation of the blade Av_1 and a tangent with the circle where the blade terminates. This tangent Ac_1 is proportional to the peripheral speed. The angle ϕ varies in practice from 15° to 90°; and in a pump with a proper volute the velocities V, instead of being equal to $\sqrt{2gH}$, are—

ϕ	=	V	
90°	=	0.83 $\sqrt{2gH}$	}
45°	=	0.94	,,
30°	=	1.03	,,
20°	=	1.89	,,
15°	=	1.355	,,

} without a diffuser or diverging delivery.

In the case of a pump with a diffuser or whirlpool chamber we take into account the inner radius and outer radius of the 4 blades r_1, r_2; and we take two ratios of these radii as examples—first, where $\frac{r_1}{r_2}=\frac{5}{4}$; second, where $\frac{r_1}{r_2}=\frac{3}{2}$.

Then—

$\frac{r_1}{r_2}$	=	$\frac{5}{4}$	=	$\frac{3}{2}$
ϕ	=	V $\sqrt{2gH}$	=	V $\sqrt{2gH}$
45°	=	0.9	=	0.885
30°	=	1.0	=	0.98
15°	=	1.33	=	1.12

Without going into all minute corrections and coefficients included in the mathematical treatment, we can indicate the calculated dimensions of a centrifugal. It may be first pointed out that, as shown in above tables, as angle ϕ decreases or increases, so does the speed. With angle = 30° and $\frac{r_1}{r_2}=\frac{5}{4}$, the speed is = $\sqrt{2gH}$; or at 15°, V = 1.33$\sqrt{2gH}$.

An angle of 90° may and has been used without decreasing the efficiency, but lowering the speed.

The head of water maintained by a centrifugal with vanes radial at the rim is found practically by $h=\frac{V^2}{g}$. h = head in feet, V = velocity of wheel rim in feet, g = 32.2 accelerating force. And not to the usual theoretical formula $h=\frac{V^2}{2g}$.

This is due to the fact that the water receives two impulses in two directions from the fan. The true centrifugal impulse is radial along the vanes, and a tangential impulse whirling it round. The velocities of the water in each direction at the moment it leaves the wheel are equal to the peripheral velocity. The resultant of the two impulses and two directions is that of the two components; hence the $h=\frac{V^2}{2g}\times 2=\frac{V^2}{g}$.

Therefore the head may be greater than that equal to $\frac{V^2}{2g}$; but owing to friction of water and obstructions to flow the calculated head is never attained in practice.

By curving the blades back the ratio between head and velocity of wheel is altered considerably, as will be seen in the tables.

The efficiency in all but the lowest lifts is above 60 per cent. And by careful design and good workmanship over 80 per cent. has been obtained in lifts of 20 to 40 feet.

The lift of a pump is limited by the highest circumferential speed at which the fan may be run, so that when the lift exceeds 40 to 50 feet it is necessary to put pumps in series. And when we require both high lifts and high values in gallons per minute we must put them in parallel.

The radius r_1 of a fan is limited by the revolutions per minute, and the

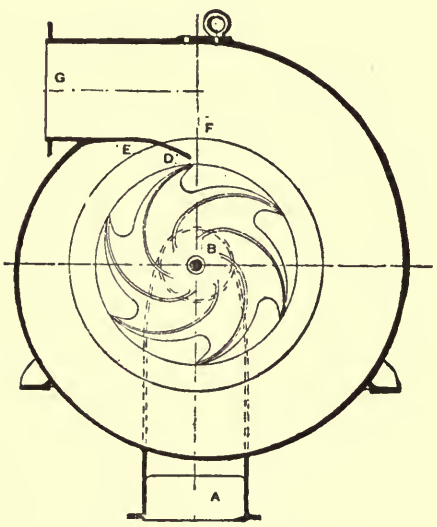


FIG. 139.—Centrifugal Pump.

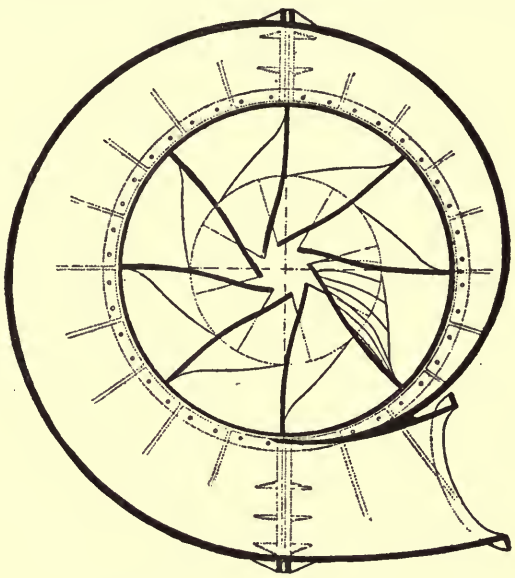


FIG. 140.—French Design of Pump.

speed in feet of the circumference of r_1. The inlet for water is limited by the value of r_2.

With a given speed both H and Q are limited, and if the limit of H is exceeded, turbines are put in series; if Q is exceeded, then they are put in parallel.

For instance, suppose a centrifugal to lift 36 feet, and V the speed in feet of the outer circumference of the fan, and that $V = \sqrt{gh} = \sqrt{32.2 \times 36} = 34$ feet; the revolutions per second are, say, 5. Then the circumference must be $\frac{34}{5} = 6.8$ feet, or 26 inches diameter. The ratio $\frac{r_1}{r_2}$ being $= \frac{3}{2}$; $r_1 = 13$ inches; $r_2 = 8.6$ inches. The velocity of inlet at the radius r_2 at the fan even as high as 22.5 feet would give a possible delivery of only 10.4 cubic feet per second as the maximum at that speed and lift; hence we would require to parallel the pump with another if more water was required, and the higher the speed the sooner the limit to the capacity of the pump is reached.

As r_2 cannot be increased without loss of efficiency, increasing the output by that means is restricted, as the efficiency falls off rapidly with decreasing values of $\frac{r_1}{r_2}$.

Fig. 139 shows an elevation in section of the usual construction, with curved pump blades. The water enters at both sides, and thus balances the pressures. The angle ϕ in this case is about 15°. Fig. 140 shows an elevation in section of the huge pumps erected in Egypt by the French, in which the angle ϕ is 90. They are slightly curved the opposite to that in Fig. 139, but not radial. A few particulars may be of interest. The shaft is vertical, and the water enters by the lower side only. The fan is much like an inward and radial flow turbine wheel, like the Hercules wheel reversed; if that wheel (Fig. 77) were driven by power and the lower end dipped into water, the water would be scooped up vertically and thrown out radially,—that is the action of these pumps. Their lift is 10 feet, and the water lifted 212 cubic feet per second; the speed is 32 revolutions per minute. The outer diameter is 12.446 feet; circumferential velocity, 20.8 feet per second $= 0.82\sqrt{2gH}$, a figure which does not at all agree with mathematical deductions; the outer case is 19 feet 8 inches diameter. The mouthpiece of the volute is 5 feet 3 inches.

The useful work done by the pump is 65 per cent. of the indicated horse-power; coal consumption, 3.85 lbs. per pump horse-power per hour.

Now, looking to this result and the results of closely reasoned and calculated pumps with volute curves and everything according to mathematical deductions, which are no better, and in many cases not so good, we come to the conclusion that the data assumed is either incomplete or in error.

Most manufacturers accept the values for ϕ and V as given above, and, having decided upon a value of ϕ for all their pumps the same, stick to it. $\frac{r_1}{r_2}$ is a ratio they also fix upon and adhere to, both being checked by careful tests on actual pumps, The velocity of influx at the inlets is usually 8 feet per second, or a little more; the velocity of discharge is from 2 to 5 feet usually per second.

The fundamental formulæ is the same as for turbines. If the water leaves the wheel without velocity, then if V is the velocity of water in feet per second the pressure due to each lb. stopped, or the pressure required to start it up to the velocity V, is $\frac{V}{g}$; and multiplying by velocity of wheel rim V_1, the useful work per lb. $= \frac{VV_1}{g}$ foot-lbs. per second. The foot-lbs. per lb. of water is $H = \frac{V^2}{2g}$, of which ηH is given to the wheel by the water or by the wheel to the water; hence $\eta H = \frac{VV_1}{g}$, the fundamental formula for centrifugal pumps and turbines, η being the efficiency factor.

In the case of the pump, η is about 0.75, and the fundamental equation H, the height to which the water is to be lifted, requires that $\frac{VV_1}{g}\eta = H$.

Without going into all the calculations, we will take—

V_1 as the speed in feet of the rim of the wheel per second.

V as the speed of the water due to head H.

r_1, the outer radius of wheel, r_2, the inner radius of wheel $= \frac{3}{2}$.

H = height to which water is to be raised, to which is added an allowance for pipe friction if pipe is long.

U = the speed of water in feet per second at inlet = 8 feet.

B = diameter of inlet.

U_1 = speed of water at outlet = 2 to 5 feet.

B_1 = diameter of outlet.

b = area of outlet of water on rim of fan at r_1, not including thickness of blades.

b_1 = area of inlet at r_2, not including thickness of blades.
Q = cubic feet of water delivered per second.
A = area of suction pipe.
A_1 = area of delivery.
N = revolutions per second.

Taking, for example, a pump to lift 32 feet and deliver 10 cubic feet per second, vanes radial at the rim.

The pump is supposed to have a volute snail delivery chamber, and either a diverging delivery pipe or a whirlpool chamber,—that is, a circular space surrounding the rim of the wheel in which the whirl of the water is reduced before it enters the snail. Let the following values be known :—

$$\left.\begin{array}{l} U = 8 \text{ feet per second inlet} \\ U_1 = 5 \text{ feet per second outlet} \\ Q = 10 \text{ cubic feet per second} \\ H = 32 \text{ feet} \\ N = 5 \text{ revolutions per second} \end{array}\right\} \text{known quantities.}$$

The peripheral speed will then be $= V\sqrt{gh} = \sqrt{32.2 \times 32} = 32$ feet per second.

The revolutions are 5 per second; hence $\frac{32}{5} = 6.4$, the circumference at r_1, or 24 inches diameter. $\therefore r_1 = 12$ inches.

$\frac{r_1}{r_2} = \frac{3}{2}$; hence $r_1 = 12$, $r_2 = 8$ inches, and V at r_2 will $= 20$ feet.

b, the breadth of the opening in the wheel at r_1, $= \dfrac{12\,Q}{V\left(\frac{2\pi \times r_1}{12}\right)}$, wherein we multiply by 12 to reduce to inches; hence

$$b = \frac{12 \times 10}{32\left(\frac{2 \times 3.14 \times 12}{12}\right)} = 0.6, \text{ and } b_1 = \frac{12 \times 10}{20\left(\frac{2 \times 3.14 \times 8}{12}\right)} = 1.4 \text{ inches.}$$

Now, to find A, usually we have the water entering on both sides, so we will divide by 2 to find A for one side. $A = \dfrac{10 \text{ cubic feet per second} \times 144}{8 \text{ feet per second} \times 2} = 90$ square inches for each side of suction pipe and side opening in the centre of turbine, $10\frac{3}{4}$ inches. To this must be added an allowance for shaft arc, which would bring the central openings to 11 inches, allowing about 8 square inches for shaft.

A_1 at discharge will be similarly $\frac{10 \times 144}{.8 \times U} = \frac{10 \times 144}{.8 \times 8} = 225$ square inches.

The suction pipe $= \frac{10 \times 144}{0.9 \times 8} = 200$ square inches.

Thus we have found the leading hydraulic dimensions. The mechanical dimensions are treated under mechanical designs of pumps.

We will now illustrate a few actual examples of recent results of tests.

The ordinary belt driven centrifugals and the direct coupled centrifugal to quick speed engines are greatly in use, especially the direct coupled, for marine circulation and bilge pumps. These are too well known to require space here.

De Laval pumps, shown in Plate IV., by Messrs. Greenwood & Batley, driven by steam turbine, are of recent introduction. The high speed enables considerable lifts to be obtained, and two pumps can be driven, the one feeding into the other in series, so that their pressures are added; and if each lifts 100 feet, the two would lift 200 feet.

They can also be joined in parallel. The following table gives some results :—

TABLE XIV.—RESULTS OF TESTS WITH DE LAVAL STEAM TURBINE PUMPS.

Type of Turbine Pump.	Revolutions per Minute.	Height of Suction in Feet.	Height of Delivery in Feet.	Quantity of Water delivered per Second Gallons.	Water Horse-power.	Brake Horse-power.	Efficiency.
50 horse-power duplex pump coupled in parallel .	1500	16.4	16.4	63.5	37.87	50.3	0.753
50 horse-power duplex pump coupled in parallel . Constructed for larger head of water than the previous .	1500	16.4	29.53	46.3	38.66	48.0	0.805
50 horse-power duplex pump coupled in series . .	2200	19.7	137.8	12.3	35.22	50.3	0.700
20 horse-power duplex pump coupled in series . .	2315	9.84	85.3	82.5	14.27	20.0	0.713

The general tendency at present is to increase the speed of everything, and it may perhaps be considered as a step in the wrong direction to reduce the speed of a machine. It may therefore be interesting to draw attention to a machine now on the market which runs at the same speed as the turbine, namely, a centrifugal pump worked direct from the turbine and at the same speed. The machine consists of a steam turbine driving one centrifugal pump direct, and another by means of gearing. The slow speed pump lifts the water and presses it into a high speed pump, which in its turn forces the water against a considerable head. The following table shows some of the results obtained by tests with one of these machines :—

TABLE XV.—RESULTS OF TRIALS WITH A DE LAVAL HIGH PRESSURE TURBINE PUMP. "TURBINE SPEED PUMP."

	Height of Delivery. Feet.	Gallons of Water delivered per Minute.	Water Horse-power.	Lbs. of Steam per Water Horse-power per Hour.	
1st Trial	312	529	50	34.2	
2nd Trial	443	629	84.4	29.0	
3rd Trial	509	529	81.6	31.3	Work for condensing is included.
4th Trial	467½	640	90.7	30.6	
		530	75.0	32.2	
		430	60.9	35.1	
		286	40.5	43.1	

In these tests the machine was mounted at a considerable distance from the boiler, and the admission steam was consequently supplied to the turbine through a considerable length of piping, which gives one reason to conclude the steam was wet. Better results

would probably, therefore, be obtained with dry admission steam, and with a higher steam pressure. The condensing water was supplied through a pipe from the slow speed pump, and the machine consequently drove its own condenser. In the figures of the table, work for condensing is therefore included.

To raise water 500 feet by centrifugal force is an achievement of no small moment.

The special characteristic of the turbine pumps, namely, their high speed—which is greatly in excess of anything previously known in connection with centrifugal pumps—is a mechanical problem which has been solved, partly by an improvement in the construction of the wheel and packing boxes of the pump, including their lubricating arrangements, and partly by paying the greatest attention to execution of all the movable parts. The high rate of speed has enabled the diameter of the pump wheel to be reduced, and consequently the passive resistance of the pump has been greatly lowered.

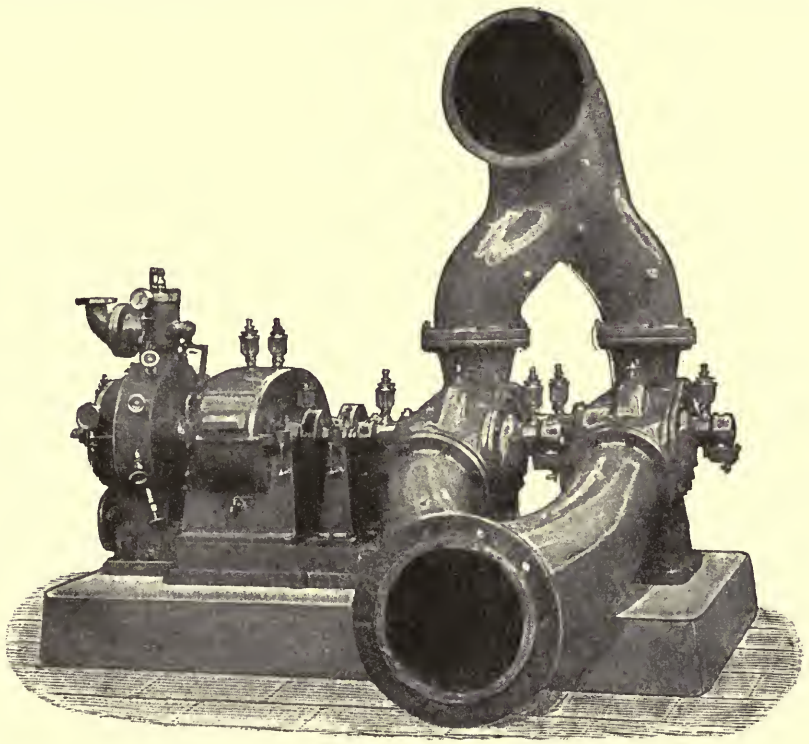


FIG. 141.—De Laval Turbine Pumps in Parallel.

This circumstance, together with the fact that no transmission belts are used, has made it possible to obtain with the turbine pumps an efficiency considerably in excess of the results generally obtained with centrifugal pumps.

Owing to the water in the pump running in a continual current, self-acting valves are avoided, and therefore the turbine pumps are very easily attended to and require very few repairs. The turbine pump runs without any shocks, and consequently air vessels are not required either for the suction pipe or for the pressure pipe.

Another great advantage of these pumps is their absolute safety against bursting. Should the pressure pipe suddenly be shut off during the time the pump is working at full speed the water pressure could not go beyond the so-called "centrifugal pressure," which, as a rule, is about 25 per cent. above the ordinary working pressure.

The dimensions of the turbine pump are small, and the weight, when compared with its capacity, is low. In these respects the De Laval patent turbine pump has the advantage over all other systems of direct acting steam pumps.

To the larger sized steam turbines, which are provided with two driving shafts, are coupled two pumps, one to each shaft. By arranging these pumps in such a manner that one of them draws the water and presses it into the other, the pressure in the delivery pipe of the latter pump is doubled. Thus a high pressure is attained, and the efficiency becomes higher than if the same pressure were to be developed by only one pump of larger diameter.

This pump thus arranged is especially well suited for a stationary fire engine for large works, or wherever the pump can be coupled to a system of piping provided with standpipes; it is also suitable as a floating fire engine, or as a mining pump.

For waterworks in small towns the high pressure pumps are particularly well adapted, in consequence of their high efficiency and their steady pressure. The stuffing

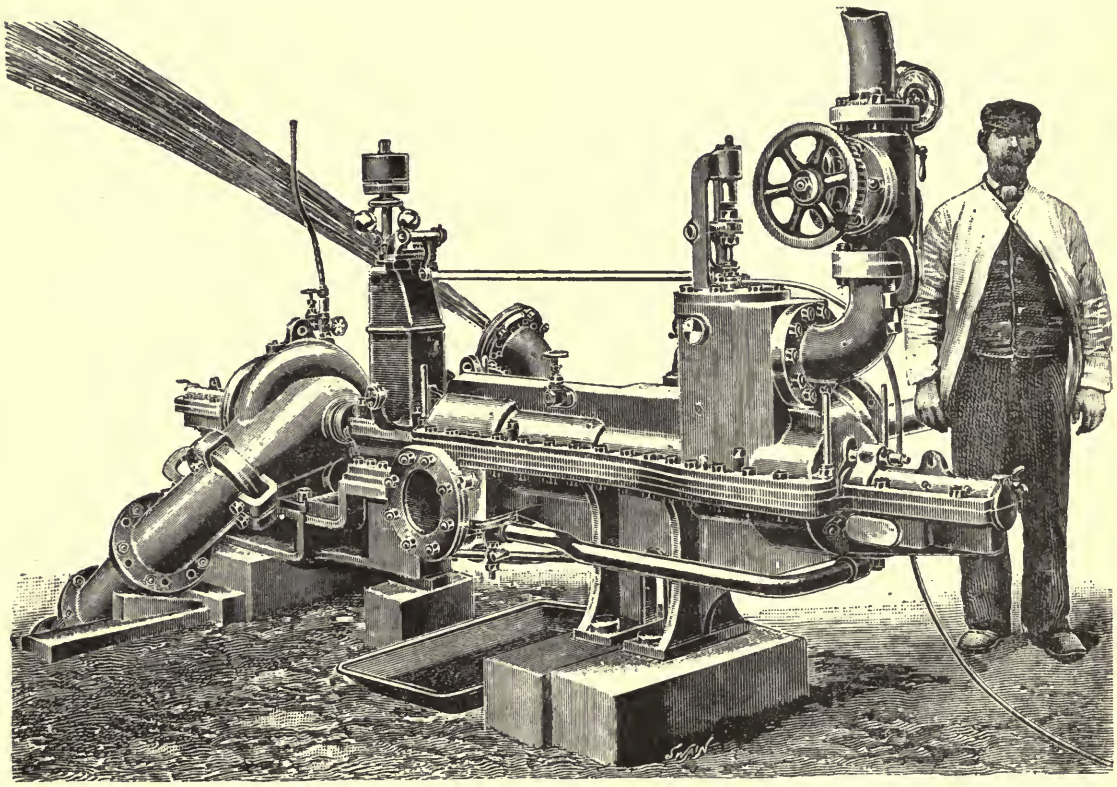


FIG. 142.—Parsons' Turbine Pump.

boxes are provided with water checks, which completely prevent the intrusion of oil into the water.

When the two pumps are coupled parallel—that is to say, when both of them draw and deliver the water at the same pressure—a most effective water raising pump is obtained, which delivers double the quantity of water delivered by the high pressure turbine pump, but only at half the pressure.

Messrs. Parsons also make high lift turbo pumps, specially designed for high speeds.

Fig. 141 represents a De Laval turbine pump with the deliveries in parallel; and Plate V. a De Laval turbine pump with deliveries in series.

Fig. 142 represents a Parsons' turbine with one pump under tests. This pump delivered 55,000 gallons per hour at 165 feet head.

Messrs. Mather & Platt, Manchester, construct pumps with fans in series inside of one casing instead of coupling separate fans.

The feature of this pump is that it consists of one or more sets of vanes, or impellers, each set running in its own chamber but upon a common shaft, the delivery pressure of the liquid varying directly as the number of chambers used. Thus, if an ordinary single pump can deliver water against a head of 30 feet, the addition of another chamber will give a final delivery head of 60 feet, while 4 chambers will enable the pump to discharge the same amount of water against a head of 120 feet.

With this multiple pump large quantities of water, to a height of as much as 150 feet, are delivered.

Recently, by still greater improvements, they are now able to deliver water against as great a head as 200 feet with a single chamber and at a high efficiency, some of the larger sizes giving out the equivalent of as much as 76 per cent. of the power put into them.

Most of these pumps run at speeds over 1000 up to 2000 or more revolutions per minute.

FAN BLOWERS AND EXHAUSTERS

These machines are centrifugal pumps for air, being designed for much higher velocities. Air being so much less in weight per cubic foot, the speeds of driving are necessarily high.

If V = velocity of the tips of a fan in feet per second, and P = pressure in lbs. per square inch, $V = \sqrt{P\,97{,}300}$, and $P = \frac{r_2}{77{,}300}$.

If Q = cubic feet of air per minute, and P = pressure in lbs. per square foot, *i.e.* the

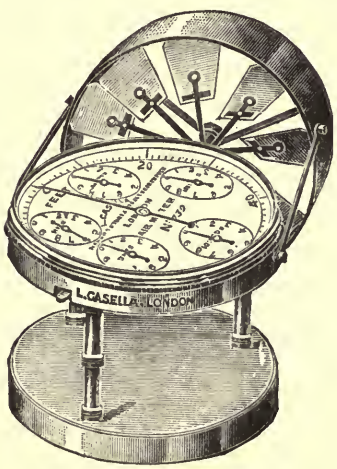


FIG. 143.—Anemometer.

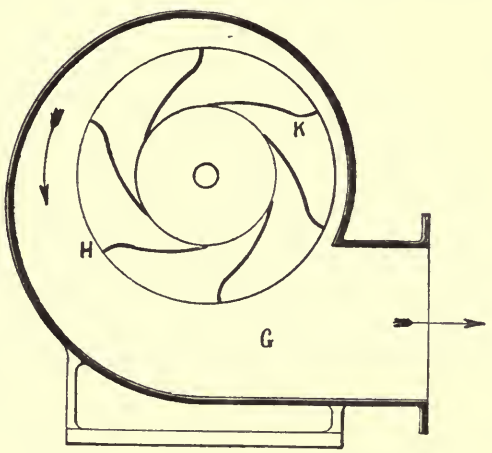


FIG. 144.—Heenan & Gilbert's Fan.

difference in pressure between inlet and outlet, then the horse-power utilised by the fan blower = $\frac{PQ}{33{,}000}$. The pressure P is measured by a manometer, a U tube, which indicates the pressure by difference of the level of water caused by the pressure. The quantity is measured by the anemometer, as shown in Fig. 143, an air meter by Casella of Holborn.

The Pitot tube is an instrument consisting of a bent glass tube on a scale. One end at right angles is fixed to the blast, the other limb is vertical. The air blast raises the column of water inside the vertical limb, and this difference of level indicates a certain velocity of flow, the depression of the column being proportional to the square of the flow.

Fans are made with all sorts of blades, and every maker claims his as the best. Some are curved backwards, some curved forwards, and others not curved at all. But as it is a centrifugal pump there is no doubt the same reasons for curving the blades as there are in that case. Professor Rankine suggested the form adopted in the Heenan & Gilbert fan shown in Fig. 144. The vanes at the inlet slice off the air and wedge it outwards, finally throwing it off at full velocity of the fan tip, which is radial. Most fans deliver the air radially, but Rateau in France and Mr. Parsons in this country have designed axial flow fans. These have the advantage of being readily constructed with fans in series, so that high pressures can be attained. Mr. Parsons in this way, by using a long series of axial flow fans with guide blades between, like his steam turbine reversed, has succeeded in producing air pressures up to 50 to 60 lbs. per square inch.

The turbine can be economically used for air or gas compression. A modified form is illustrated here attached to an ordinary steam turbine (Fig. 145); this particular set is capable of developing from 300 to 400 horse-power, and of compressing air to 25 lbs. per square inch. Compressor can be designed for practically unlimited pressure by coupling two or more in tandem with or without intermediate coolers, and where electricity is available they may as readily be motor driven.

Large steam sets are especially applicable for blast furnace work; there are no air valves to damage by heat, and, as is usually necessary, the pressure can be increased by 50 per cent. temporarily by means of a byepass valve on the steam engine admitting high pressure steam to the low pressure parts of the turbine.

In rock drilling or mining work, motor driven compressors can be attached to small cars, and the compressors taken right up to the face The conveyance of an equal amount of power electrically over long distances is much less costly than by pipes from large compressors at bank.

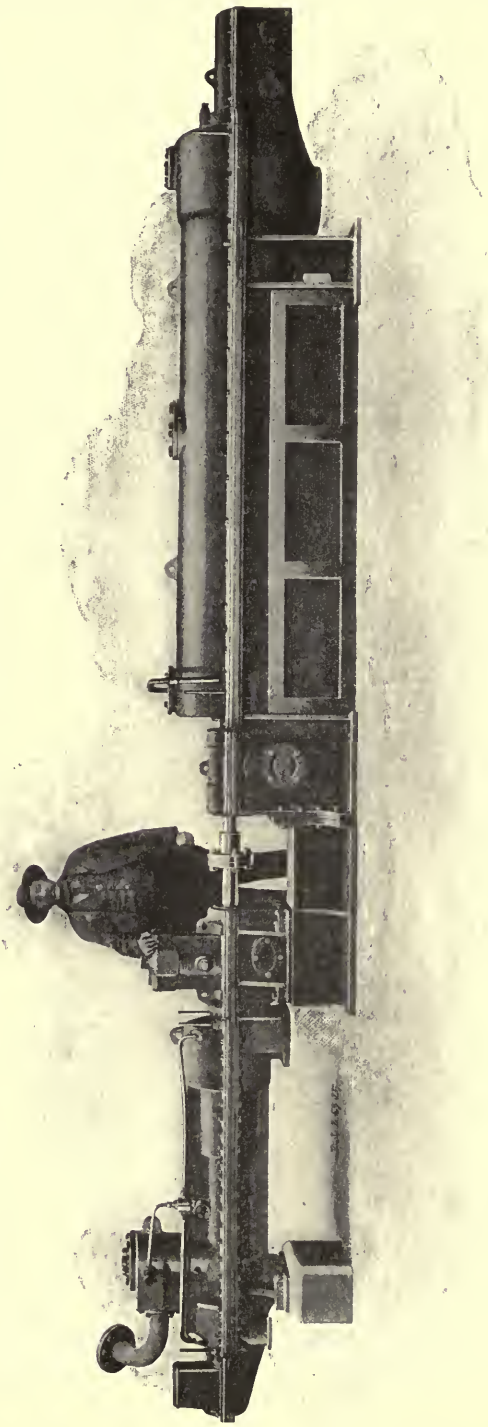


Fig. 145.—Parsons' Turbine Air Compressor.

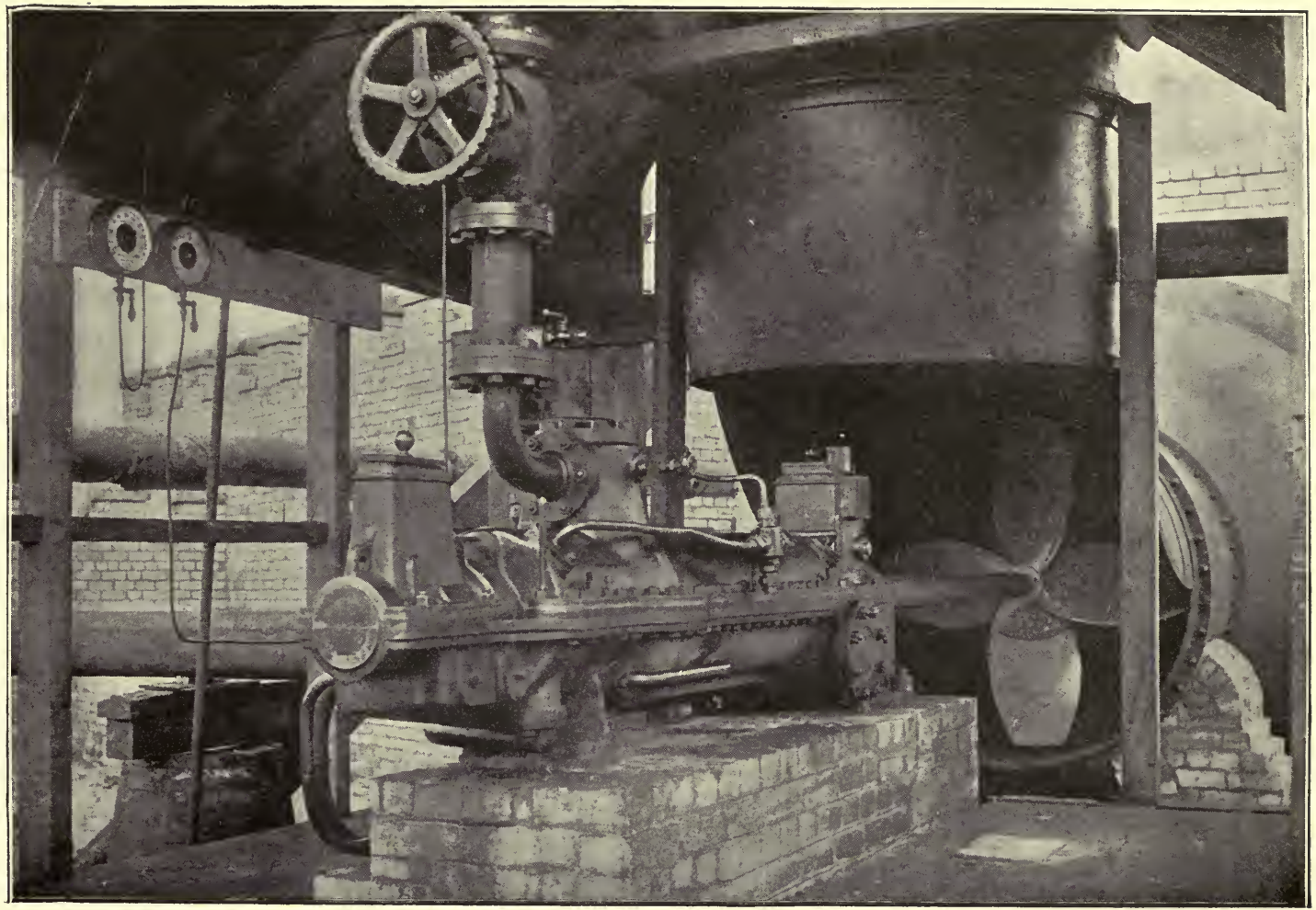


FIG. 146.—Parsons' Screw Mine Ventilating Turbine and Fan.

The weight of the set shown in the print is about 7 tons complete.

The Rateau fan wheel for axial flow is shown in Fig. 147. A number of these can be threaded on one shaft driven at a high speed and with guide passages between, so that the pressures are added together at the end outlet. And as the volume of the air diminishes as it rises in pressure, succeeding wheels are made of small diameter and with fewer blades.

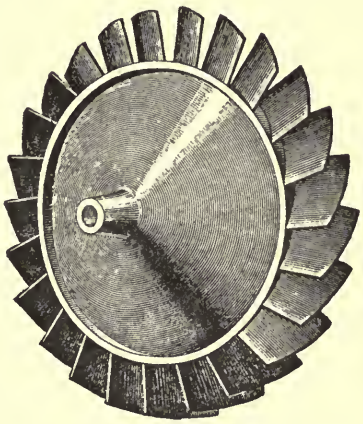


FIG. 147.—Rateau Fan.

Mr. Parsons' wheels are like his turbine wheels, with curved blades.

The Rateau fan lends itself very nicely to electric driving.

The great usefulness of fans in ventilating mines is their most important work, very large and powerful fans being required either to force air down or draw air up through the working, to clear out gases, foul air, smoke, steam, etc., and supply fresh air for the miners.

At Clara Vale Colliery a screw fan driven by a steam turbine has been fitted. It is illustrated in Fig. 146, and has considerable interest as a more efficient and reliable form of fan than the centrifugal form.

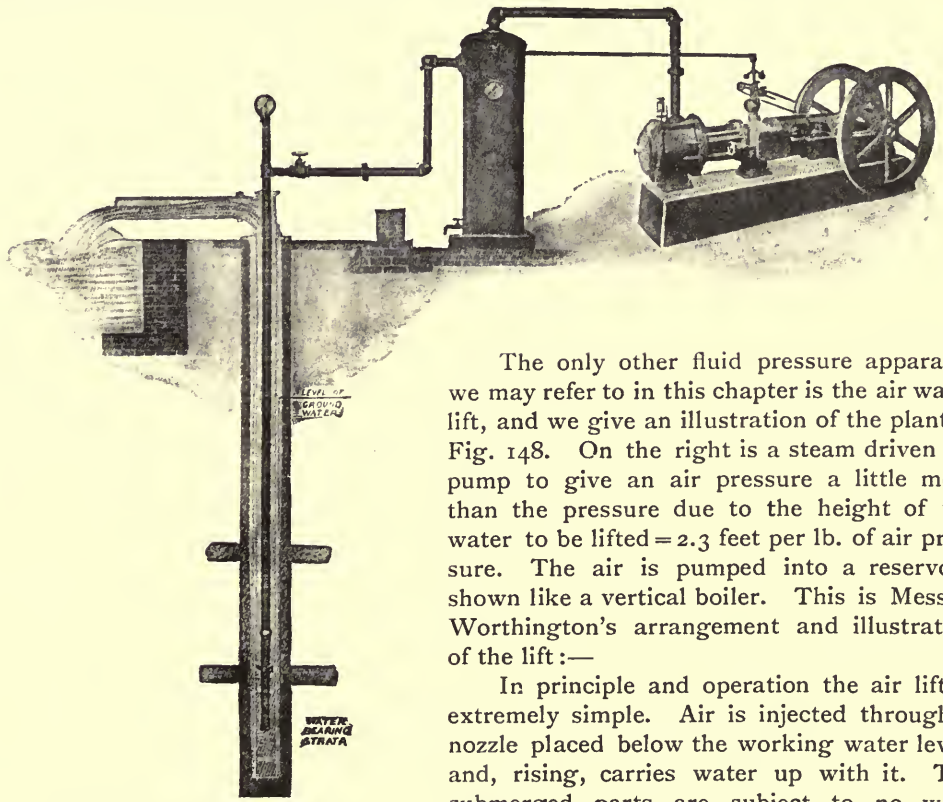


FIG. 148.—Air Water Lifter.

The only other fluid pressure apparatus we may refer to in this chapter is the air water lift, and we give an illustration of the plant in Fig. 148. On the right is a steam driven air pump to give an air pressure a little more than the pressure due to the height of the water to be lifted = 2.3 feet per lb. of air pressure. The air is pumped into a reservoir, shown like a vertical boiler. This is Messrs. Worthington's arrangement and illustration of the lift:—

In principle and operation the air lift is extremely simple. Air is injected through a nozzle placed below the working water level, and, rising, carries water up with it. The submerged parts are subject to no wear whatever, and, except for corrosion, are practically indestructible, even when working in the gritty water so fatal to deep well pump pistons. The air compressor and receiver, the only parts needing attention, are at the surface, and readily accessible. When proper submergence of the air nozzle can be

assured the yield of a well previously fitted with a deep well pump is doubled or trebled by the use of the air lift. The sparkle and life imparted to the water raised, and the cooling effect of the expanding air, make this method peculiarly suitable for waterworks purposes; while, owing to the absence of pump rod and plunger friction and the superior steam economy of the fly-wheel compressor, better duty can be obtained than with the old style pump. When warm water is to be raised, its heat, instead of being a disadvantage, is a source of increased economy.

A certain minimum ratio between submergence of the nozzle and height of lift is, however, essential to the successful working of an air lift.

This concludes a review of wind and water prime movers, with some notice of machines working on hydraulic and pneumatic principles, which are not strictly speaking prime movers, such as injectors, centrifugals, fans, and fluid-on-fluid pumps. Yet, as they play a not unimportant part in prime movers' applications and operations, they are worthy of some attention.

The hydraulic jet has been examined carefully and fully, for it is more than likely to play a very important part in marine propulsion, as it offers a solution of the difficulties of high speed, driving engines direct coupled to screw propellers outside of the vessel. The jet presents some difficulties also, due to the fact that high pressure jets cannot be used with economy, as the speed of the jet must not exceed the speed of the vessel by more than twice. This entails using large quantities of water, the chief objection being the added weight of the water inside the vessel.

On the other hand, we have seen the many advantages the jet has over the screw.

CHAPTER III

THE STEAM TURBINE

THIS prime mover, long the dream of advanced and leading scientific engineers, has at last reached the stage of perfection at which it can with confidence be employed to replace the reciprocating piston and cylinder engine of any but the smallest size. The bigger and more powerful the steam turbine, the better it compares with the piston and cylinder engine if condensing is used. And even non-condensing Parsons' turbines compare well with piston engines non-condensing.

Steam turbines are either of the pure impulse type, like a Pelton wheel, or of the pressure type like the pressure turbine, of which we may cite the vortex wheel; it will be observed that the so-called pressure turbines, are only partially pressure turbines, for most of them have curved guide blades in order to obtain also the impulse due to change of motion. The De Laval turbine is purely an impulse turbine, the steam being thrown at the highest possible velocity on the curved blades of the wheel, where its kinetic energy is converted on the blades $= \frac{V^2}{2g}$ per lb. of steam.

In Parsons' turbine the steam acts by its velocity being arrested on the movable blades in the same way, and also by its pressure reacting as it leaves the movable blades. Flowing from one wheel to the next, its velocity is gradually absorbed in each, and it expands from one into the other wheel. The kinetic energy $= \frac{mV^2}{2g}$ taken for each wheel. There is the third turbine, a purely pressure turbine; the original of which was Hero's turbine, in which the motion and power is due entirely to unbalanced pressure. The first patent for this type was granted in 1784 to an engineer with the name of Wolfgang de Kempelen. As a matter of fact, there is no material difference between the impulse and reaction turbines, so that only two types may be considered.

As the steam turbine has come to be a practical prime mover of great importance, we may briefly review its rise and progress. That it will be much improved upon in construction is beyond a doubt, and some of the earlier proposals may still come forward and claim attention. It is as well to know what they are or were.

To consider them in their order of time would be of no service. We will consider them as a class at a time, beginning with the Hero turbine, which may yet be of service. Presently the difficulty with this type is the high speed required of the wheel to obtain anything like efficiency; there is also the difficulty of getting a steam-tight joint at the admission orifice for the steam into the wheel at very high speeds, and a third difficulty is to get a wheel made strong enough to stand the strains of centrifugal force at these high speeds.

Fig. 149 illustrates the Hero engine in its simplest form. Little toys in this form were common at one time, made of glass. A brass or copper hollow ball, with a neck

like a door knob, has two arms at right angle to the axis attached to the neck; it is pivoted at the bottom, and guided at the top by a stem. On applying a flame to the bottom, steam is generated, which in passing out of the nozzles leaves an unbalanced pressure which drives them backwards in the opposite direction to the steam issue. This plan of revolving the boiler with the nozzles obviates any difficulty with steam joints.

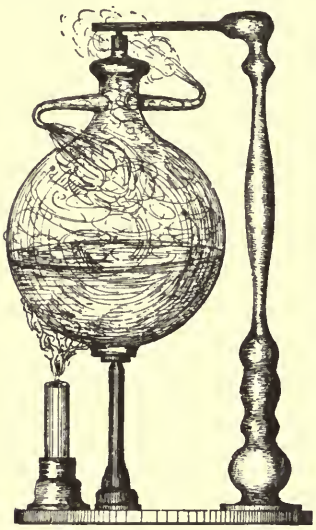


FIG. 149.—Hero's Engine. (200 B.C.).

De Kempelen's engine was the first practical attempt to apply this form of turbine to actual work. It is shown in Fig. 150, wherein A is the boiler, with a stop valve C, a safety valve B, and a vertical pipe carrying a horizontal pipe E, which is free to revolve; it having a sleeve joint as shown in Fig. 151, whereby it can move freely and yet be steam-tight.

At the extremities of the horizontal tube a hole is made, shown at DD, through which the steam escapes. As the inside of the pipe opposite the holes has a larger area by the size of the hole than the side with the hole in it, the pipe will be pushed backwards by the steam. Thus if the size of the hole is 1 square inch, and the pressure 10 lbs. per square inch, then there will be an unbalanced pressure of 10 lbs. on the side of the pipe opposite the hole, which will propel the pipe in that direction away from the hole.

Many writers describe this action as "reaction"; but it is not reaction, but direct pressure which drives the Hero turbine.

The discs on the end are weights for fly-wheel purposes. It will be evident that

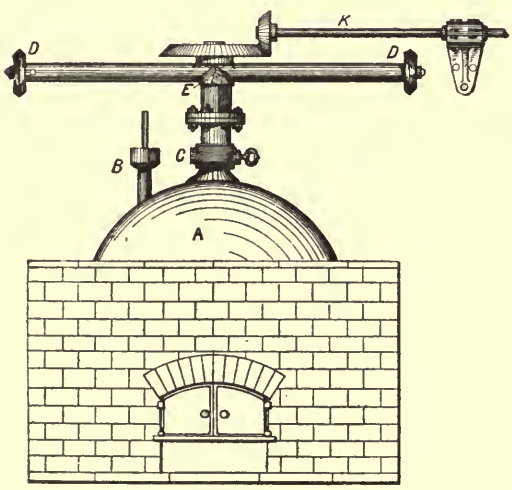


FIG. 150.—De Kempelen's Engine.

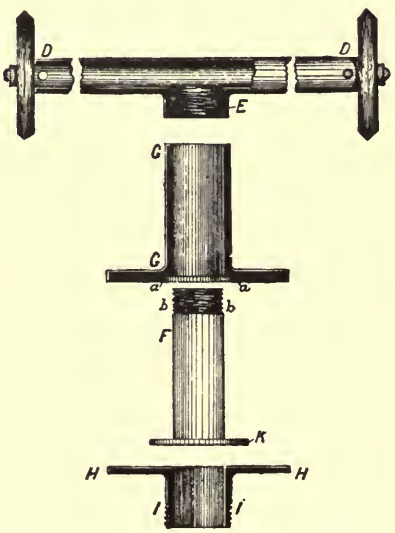


FIG. 151.—Details of Steam-Tight Joint of De Kempelen's Engine.

such an arrangement would be very limited in speed. The pipe would soon reach its breaking strain as the speed increased.

The next improvement was made later on in 1849 by Mr. Nasmyth of Patricroft. It is thus described in the *Practical Mechanics Journal*, vol. i., 1849:—

Fig. 152 is a complete longitudinal section of the steam wheel, with its frame, saw, and foundation plate; and Fig. 153 is a transverse section through the wheel

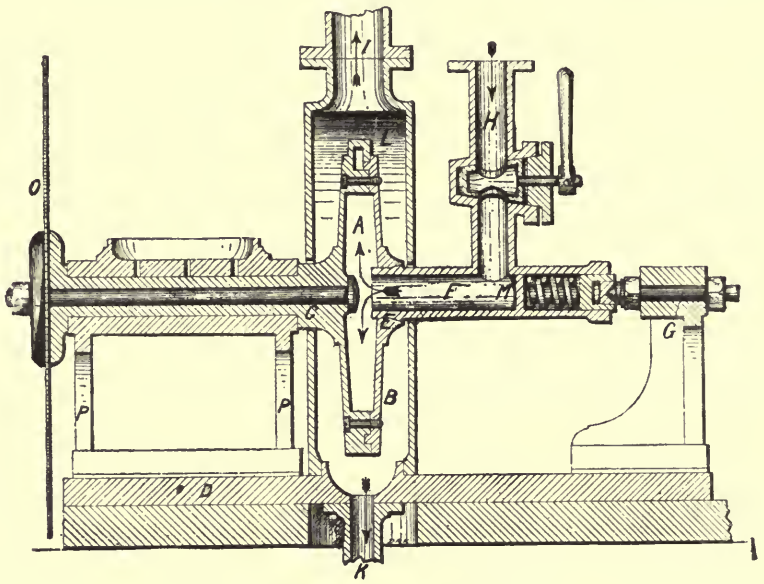


Fig. 152.—Nasmyth's Turbine Saw.

and waste steam case. The wheel A is a hollow open disc, with a loose side B checked into it, and bolted down by nine bolts. The hollow saw shaft C is cast in one piece with the wheel, and is carried in the long bearing of the double pedestal PP,

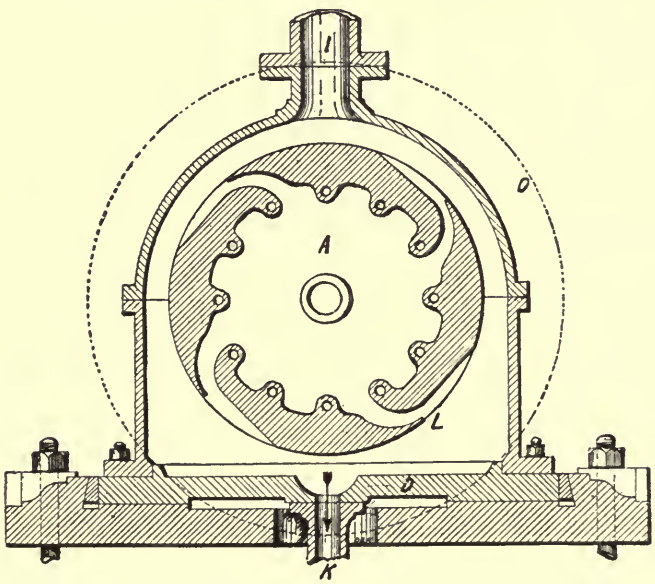


Fig. 153.—Nasmyth's Turbine Saw, showing Nozzles.

which is bolted down to the foundation plate D. The contrary side of the steam wheel is steadied upon the conical bearing E, on the end of the supplying steam pipe F, which rests between the adjustable centre G of the back pedestal and the conical

aperture in the centre of the side plate B of the wheel. The arrows show the passage of the steam into the wheel by the pipe H; and the escape of the waste steam and condensed water by the upper and lower pipes I and K, connected with the outer casing of the wheel L. A compensating spring is cleverly arranged in a prolongation of the steam pipe F to counterbalance the pressure of the steam upon the area of its ingress opening at M. A helical spring N is placed in the open end of the pipe, so as to abut against the closed end of the pipe M. This spring is acted upon by a short sliding block, urged forward by the adjusting centre G; in this way it is compressed to such an extent that its reaction will just compensate for the steam pressure upon the supplying area, and so prevent the possibility of undue friction upon the conical joint of the side plate of the steam wheel.

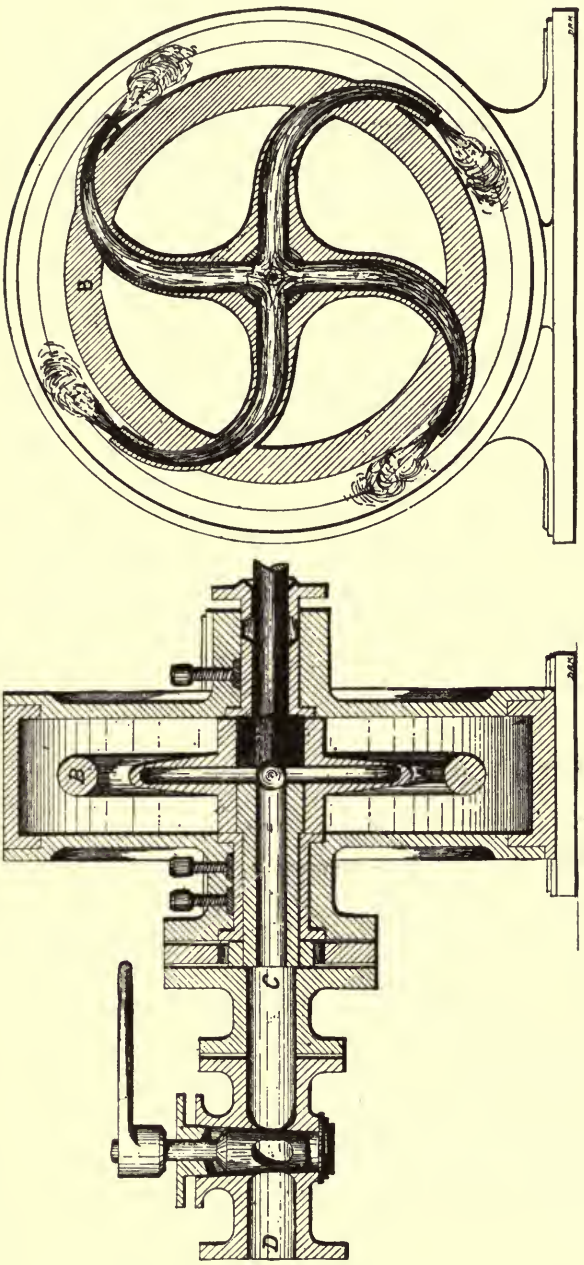


FIG. 154.—Steam Turbine Fan, 1854.

The waste steam case L is cast in two halves, bolted together at the centre by means of side lugs, the whole being firmly bolted by flanges at the bottom to the base plate, which has a steadying rib cast on it to fit the interior of the lower edge of the case. The total area of the four steam apertures is 1½ square inch; steam pressure, 60 lbs.; revolutions per minute, upwards of 2000. The time taken to cut a bar of the average section is 10 seconds, and it is to this limited period of action that this species of steam propeller owes its peculiar fitness. For although the wheel has in reality very little power, yet its great speed admits of a large accumulation of momentum between each sawing action, and practically gives it ample power for the 10 seconds duration of work.

Another wheel designed to run a fan blower was described in the same volume. It is shown in Fig. 154 in section. The steam enters the centre of the wheel and escapes by curved arms, as shown. The longitudinal section shows the hollow shaft through which the steam enters the wheel.

It is evident that in both these the idea of the design is to obtain a wheel which

could be run at high speeds. Unfortunately, in neither case was the diameter of the wheels stated; but 2000 revolutions is given as the speed.

It was pointed out at the time that the speed of the wheel would require to be some large fraction of the speed of issue of the steam from the nozzles to get any efficiency, and that speed was calculated at 1200 per second.

About that time much attention was paid to steam turbines. Pilbrow, the great turbine inventor, had made his experiments and patents known.

De Kempelen had evidently seen the impossibility of obtaining efficiency with steam, and proposes, but unfortunately without illustrations, a method whereby he uses the steam to propel water alternately from two vessels. Through his turbine, he says, the steam is to be admitted alternately from the boiler to the two vessels, and presses upon the surface of the water, forces it into the turbine; the water is to be returned to the receivers and worked hot.

In the same year James Watt patented this same idea. It is shown in Figs. 155 and 156 in diagram. A vessel A, B, D, E, C is mounted upon a pivot J, and supported by collar K at upper end. The vessel is divided into two, with an opening HH in each at the side for the escape of the fluid, and there are two clack valves F and G for the entry of the fluid, which surrounds the rotating vessels up to the level of the overflow O. Steam is supplied by pipe L alternately to the two divisions of the vessel, so that when one is filling up with fluid the other is being emptied by the steam pressure.

Watt proposed to use mercury, oil, or water as the acting fluid.

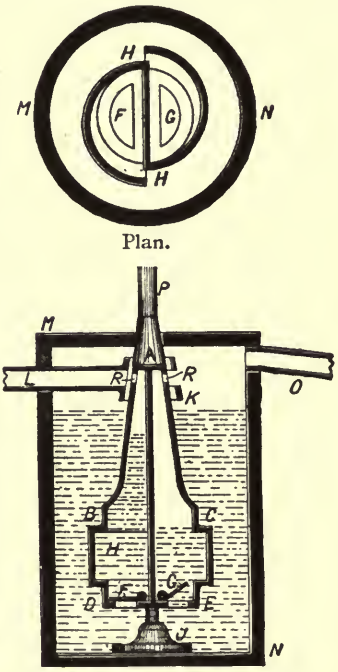


FIG. 155.—James Watt's Turbine.

Both these proposals when examined show that they had hit on the idea of the pulsometer in a primitive form. In point of fact, a pulsometer pivoted at the bottom, and with its water inlet under water level and its steam inlet in a stuffing box free to revolve, would constitute a modern and more effective turbine on the Kempelen and James Watt plans for fluid turbines. Instead of the ordinary outlet there would be two nozzles, one on each vessel for the ejection of the water at a tangent to the vessels.

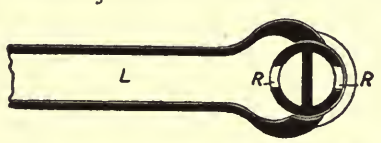


FIG. 156.—Steam Inlet of Watt's Turbine.

Trevithick's patent describes Hero's turbine without much improvement upon Hero's machine.

Ericsson, in his patent of 1830, shows considerable improvement upon Hero, as will be seen from the two figures—one a cross section, the other a view of the wheel. He forms conical nozzles either on the side of the wheel, or on the outer periphery as shown at rr_1, Fig. 157.

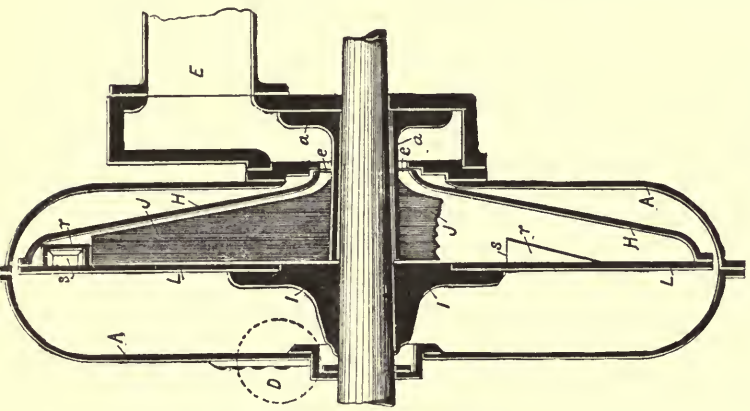
The fixed vanes J are carried on a fixed sleeve *a* on the shaft, which moves easily within the sleeve. The steam enters the nozzles from the outside or inside, but the fixed vanes are always placed so as to prevent the rotation of the steam.

These fixed vanes reveal the fact that Ericsson had found it necessary to stop the rotation of the steam against a fixture. The author of this book made some experiments with a similar wheel designed for high speeds in 1890, and found that without a set of fixed vanes in the case where the steam enters from the outside and escapes by the axis it was not possible to get it to work, for the steam whirled round and round until, by its own friction against the wheel, it lost its energy and escaped,

and the friction of the steam balanced the pressure on the jet, and there was no motion.

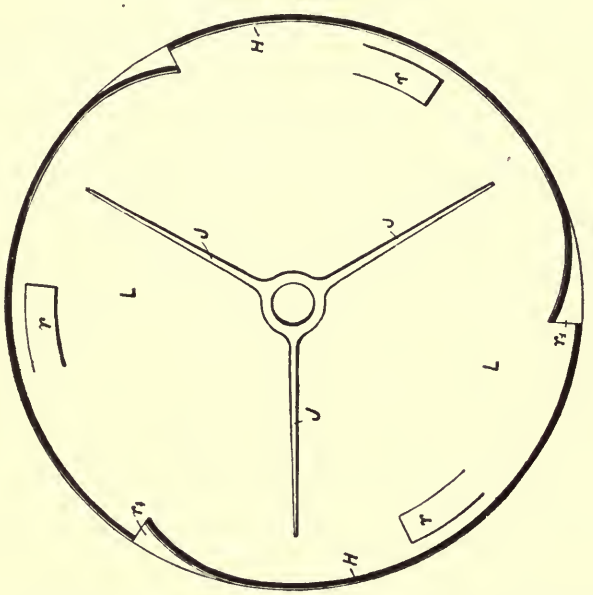
Alexander Morton, of Glasgow, the inventor of the ejector condenser, also found this out, for in his first patents he describes a steam turbine on Hero's plan, in which he used a series of concentric wheels, somewhat like Ericsson's wheel (Ericsson, by the way, was the inventor of the screw propeller). The idea is that the steam entering the inner one will propel it by pressure on the nozzles, then expanding, do the same in the next outer ring, and in the next. It was found, however, on trial of this series of wheels, that the effect was nothing; the steam simply expended its energy in rushing round in the opposite direction to that which the wheels ought to have rotated, and its friction counterbalanced the steam pressure on the outflow nozzles. In his next patent we find fixed blades inserted to stop the rotation of the steam. There are three wheels concentric with diverging nozzles on their periphery, and between these there are fixed blades and an outer row of fixed blades on the casing; these stop the whirl of the steam.

Nozzle of Ericsson's Turbine.



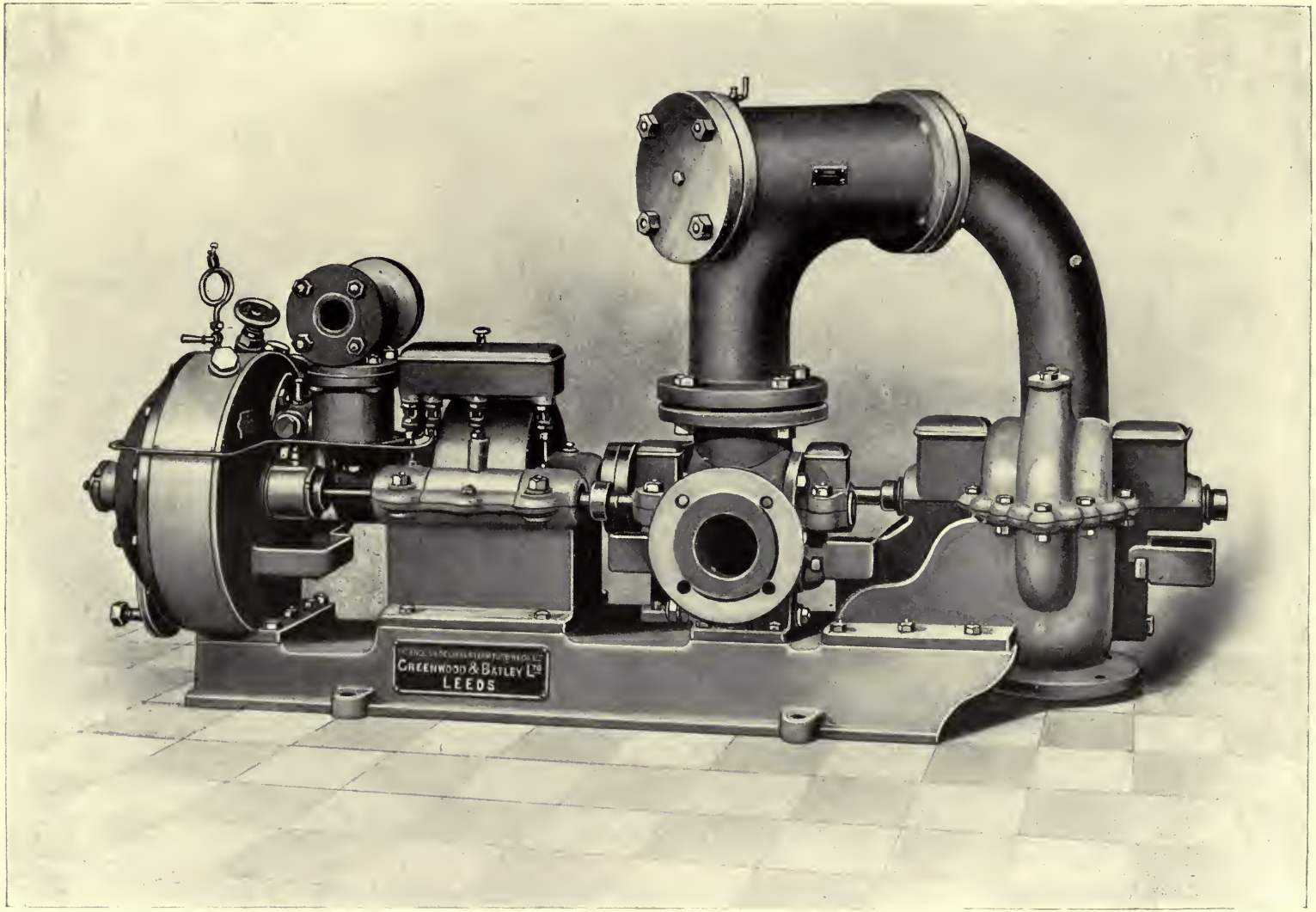
Longitudinal Section.

Fig. 157.—Ericsson's Turbine.



Ericsson's Wheel.

On examination of these early machines it will be seen that although capable inventors, from Watt's time forward, have tried to improve upon Hero, they have made very little progress. Von Rathen made Hero turbines with diverging nozzles; Ericsson and Morton also employed diverging nozzles, but a little reflection will show that the gain to be obtained by diverging nozzles in this type of turbine is small—in fact, nil. All of these inventors failed to grasp the fact that the improvements required were such as would enable the wheel to run at enormous velocities at the steam orifice. We will now take Parsons' design of pressure turbine, Fig. 158; it is a reaction engine, consisting of one or more pairs of arms mounted upon a spindle, and so arranged that steam is admitted to the interior of the said arms by suitable passages in the spindle, and is discharged from each pair by apertures at the ends of the arms arranged to discharge at or near the tangent to the circle swept by the arms.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The arms are arranged within a hollow case or series of cases. In one construction (Fig. 158) a series of circular cases D, D^1, D^2 check into each other circumferentially, and form one larger case divided by partitions into a number of chambers. One spindle passes through the series of chambers and carries upon it a pair of arms in each chamber. One arm, however, may be carried in each chamber provided with a suitable balance piece or weight; or any number of arms may be so carried, provided that balancing is carefully allowed for. One end of the spindle passes through a high pressure steam chest B, and a number of grooves turned in the spindle work in a bearing having corresponding grooves; one set of such grooves E prevents the high pressure steam from leaking into the atmosphere from the said high pressure steam chest, and another set of grooves E^1 prevents the high pressure steam from leaking into the first chamber. Between the two sets of grooves and the spindle radial apertures C^1 are drilled to a hollow, bored out or formed in the spindle. Similar apertures C^2 lead from the said hollow to the interior of the boss off the first pair of arms; the arms A, A^1, A^2 are cast hollow with a passage of uniform area and of flattened section. The exterior is also of a flattened section sharpened towards both edges in order to provide arms capable of passing rapidly through the steam in the casing without creating much resistance. That is, the arms are arranged to move on edge through the steam within the casing, and the necessary section is provided by making them sufficiently wide. These arms rotate at about 6000 revolutions per minute, and hence it is necessary that there should be a considerable margin of strength to resist the great centrifugal force. To provide this strength the exterior figure of each arm is conical, that is, the arm is thicker and broader at the root near the boss than it is at the extremity. Slots a, a^1, a^2 are cut in the edges of the arms to furnish discharge orifices and permit the steam by its reaction to propel the arms and hence the spindle. The steam passing from these orifices fills one chamber of the casing and passes thus through the boss of the next arm, which boss is turned to fit a bored opening in the wall of the chamber. A series of passages through the said boss permit steam to flow from the first chamber into the interior of the second pair of arms; thus the steam discharges by emission apertures into the second chamber of the casing. From the second chamber the steam passes similarly through the boss of the third pair of arms, and by emission jets to the third casing, whence the steam may pass to the fourth pair of arms and chamber, and so on.

As the steam passes in this manner from chamber to chamber through each pair of arms the pressure falls, and each pair by reaction adds to the energy of rotation of the steam wheel or turbine spindle. The interior of the passages and emission apertures of the arms are arranged in continually increasing area in order to allow for expansion.

It has been proved that steam cannot flow at more than 1440 feet per second from an orifice, and that an expansion of 1 to 1.63 is sufficient to gain this velocity; therefore in a compound turbine of the reaction jet type, when working with 100 lbs. steam and exhausting into the atmosphere, it is necessary to employ at least five sets of arms in separate cases, thus limiting the expansion in each issue to under the above-mentioned ratio of expansion. For this reason reaction wheels like Hero's engine, employing one wheel only, have not given good results.

The boss of each arm projecting through the partitions is arranged as a nice fit, and grooves or serrations may be turned in the bored apertures in the chamber walls; a groove packing is used for resisting high steam pressure. Such a packing is described in Specification of Patent, No. 1120 of 1890. To return to the turbine, we shall quote only two of the claims:—

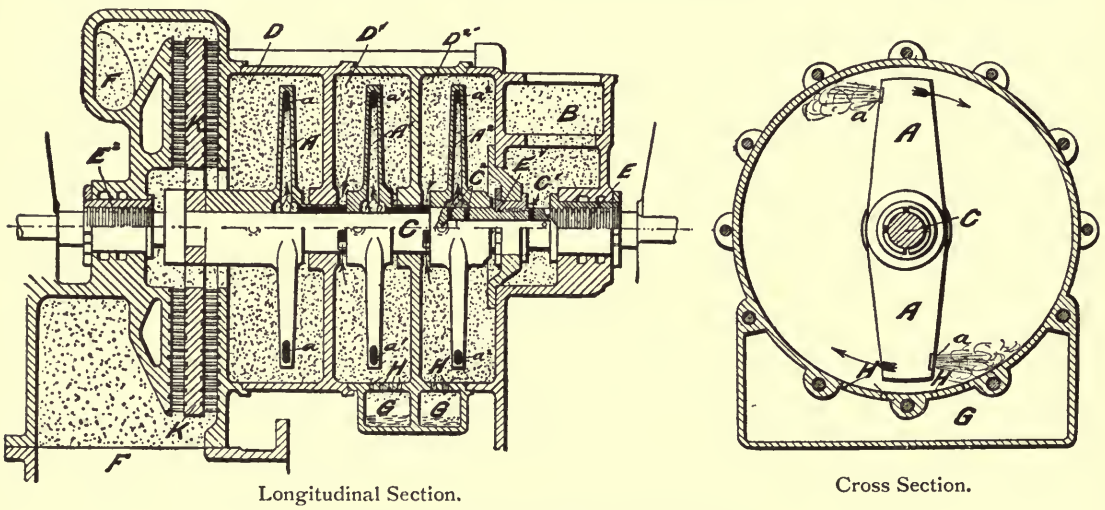
1. A steam wheel or engine consisting of a number of hollow reaction arms arranged in sequence in a number of chambers, the said arms having discharge apertures of increasing area from the high pressure to the low pressure end.

2. A steam wheel or engine comprising a series of chambers in which reaction arms are caused to rotate, the said chambers being provided with drainage apertures HH, and drainage boxes or spaces GG, substantially as hereinbefore described.

A speed of 1400 feet per second at the periphery is necessary for the full efficiency, whereas none of these wheels could with safety run above 100 per second in peripheral speed in feet.

The Hero engine in the form shown in Fig. 153, but designed for a speed of about 30,000 feet per minute at the periphery, would be as efficient as the De Laval or Parsons' turbines without any special nozzles.

We may pass over all the abortive attempts of many inventors to make a successful Hero turbine. Until 1893 we find only Mr. Parsons' patent for a steam turbine on Hero's method. It is shown in Fig. 158, a longitudinal and cross section. Pilbrow, fifty years before, had shown that by putting turbine wheels in series a greater economy



Longitudinal Section. Cross Section.

FIG. 158.—Parsons' Turbine—Hero Type.

could be effected, for the difference of pressures could be reduced, and the velocities also to some extent reduced.

Mr. Parsons in this specification calls the wheels reaction engines; strictly speaking, they are purely direct pressure wheels.

The reasoning regarding the expansion from 100 lbs. to atmospheric pressure in five stages—that is, in a series of five wheels—is correct enough; but while that arrangement would be more economical in steam than one wheel, the necessity for running the wheels at some large fraction of 1440 feet per second still remains.

This turbine, in order to run at a much lower speed than 1400 feet per second, would require not five wheels in series, but fifty, as the speed and pressure of the steam would have to fall by very small steps to get the small velocity of flow.

The velocity is equal to $64.4\sqrt{x}$ for air with a density of 0.080728 lbs. per cubic foot, where x = inches of water required to balance the difference of pressure $p - p_1 = p_2$. If we were working with compressed air a difference of $p = 100 - p_1 = 92$, $p_2 = 8$ lbs., the water pressure column to balance 1 lb. = 2.24 feet, or about 27 inches of water.

To calculate for steam, we take the volume of steam at the given pressure, and thus if the difference of pressure between the first wheel and the next were $100 - 93 = 7$, or nearly 0.5 of an atmosphere, at 100 lbs. pressure, the volume of a cubic foot of steam

is 272 times the volume of water; hence we get $V = 8\sqrt{272 \times 0.5 \times 32} = 528$ feet per second. We would require to calculate each step separately as the pressure dropped.

It is thus seen that to reduce the speed to 500 per second, or 30,000 feet per minute, would require a series of wheels in which the steam expanded by a drop of less than 7 lbs. between wheel and wheel at the start, and a less and less drop towards the exhaust for the volume of steam at atmospheric pressure equals nearly 1800 times that of water.

The greatest fall allowable is at the high pressure end of this series, and that is 7 lbs. in this case; so that it is quite clear that if any efficiency is to be got out of a series of Hero wheels their number must be great, and the difference between each in pressure small; for the speed in feet could in few cases be allowed to reach even 500 per second. Mr. Parsons proposes to run the wheels at 6000 revolutions per minute. Now, if we allow an efficiency of, say, half, $\frac{528}{2} = 264$ feet per second, or 15,840 per minute; then $\frac{15,840}{6,000} = 2.64$ feet would be the circumference of the circle described by the wheels at the diameter of the orifices. This diameter would then be about 10 inches.

The difficulties in the way of making a turbine successfully on this plan are

Fig. 159.—Double Wheels. Hero-De Laval.

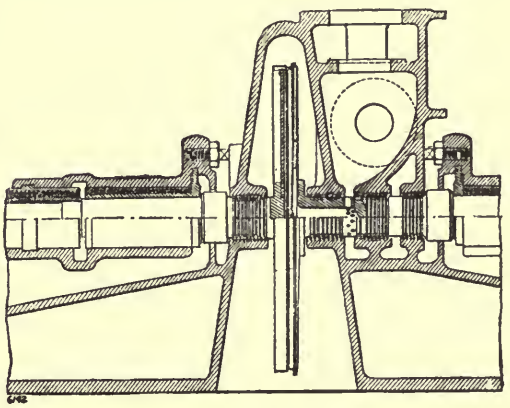


Fig. 160.—Double Wheels. Hero-De Laval.

many, and no proposal up to the present time has done much, if anything, to overcome them.

Working with an inexpansible fluid like water, it makes an excellent turbine; the difficulties begin when a fluid which must be worked expansively is to be used.

The best that can be done with it at present is to compound a Hero wheel with a purely impulse wheel, that is, to combine a Hero wheel with a De Laval wheel, and thus reduce the revolutions by one-half. This idea occurred to the author some two years ago, and has since been patented by Mr. Parsons.

A diagram showing the principles of this turbine is shown in Fig. 159. The inner wheel is a Hero wheel with diverging nozzles, in which the steam acquires its maximum velocity before striking the outer wheel, with plain curved blades like the De Laval wheel.

If we take a De Laval wheel and place it face to face with a Hero wheel in which the noozles are made diverging as in a De Laval wheel, the two wheels when steam is supplied to the Hero wheel will revolve in opposite directions with about equal torque, so that they may be geared to one common shaft, or each may drive a separate dynamo.

The turbine of Mr. Parsons combines two wheels in this way.

Mr. Parsons' patent specification describes his arrangements of the double wheel type (Fig. 160) as follow:—

In fluid pressure turbines of the De Laval type the method of securing a high relative velocity between jet and bucket with reduced skin frictional losses, by rotating in opposite directions the element-carrying nozzles and the element-carrying buckets or vanes against which the fluid impinges, is substantially as described. The arrangement is preferably such that the working fluid, after impinging on the vanes, passes to the exhaust without interfering with the action of succeeding jets. The shafts of the counter-rotating elements may be co-axial, and may respectively carry the two reacting parts of a dynamo machine. Reversing turbines of this class may comprise a separate set of nozzles (which may be fixed), supplied from a separate pressure chest, and adapted to direct working fluid against the reverse sides of the buckets, or against a row of reversely set buckets. The inventor says: "By my new method I produce a steam turbine of remarkably simple construction, which in operation allows great reduction in the losses from the skin frictional resistance; and this renders the single-expansion type for the first time admissible for direct coupling to dynamos or screw propellers or other purposes, where the speed of revolution is moderate. Apart from power saved in gearing dispensed with, such turbines have only one-fourth the disc stresses and one-fifth the skin resistance of the simple form."

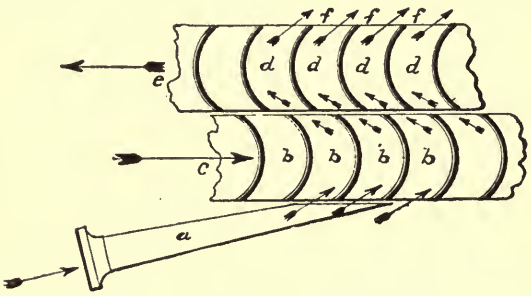


FIG. 161.—Pilbrow's Double Turbine.

This principle of construction I have carefully investigated, and my conclusions are, that it introduces insurmountable mechanical difficulties. It only reduces the revolutions by one-half, while it at the same time introduces a hollow wheel with steam inlet around the shaft, subject either to much leakage or great friction.

Pilbrow again anticipated this placing of two wheels together in order to get smaller speeds or a "high relative velocity," and his plan is shown in Fig. 161, in diagram. The steam jet enters the first wheel and propels it in one direction, it then enters the second wheel at full velocity and propels it in the opposite direction, and the two wheels can be geared together so that they add their power to one common shaft. With the knowledge given by the De Laval specification regarding diverging nozzles we would now make the two wheels as shown in diagram, Fig. 162. In this turbine the steam enters by a converging jet into a wheel with diverging passages, so that there are no difficulties about the steam inlet as there is with a hollow wheel. The steam in passing through the first wheel propels it by unbalanced pressure, and also expands to atmospheric pressure, and strikes the second wheel with full velocity as in the De Laval wheel; we thus get the double effect. In this turbine the two wheels are face to face and overhung on the end of shafts, so that they can have some free lateral motion, and thus find their own centre of gravity, around which they revolve with silence.

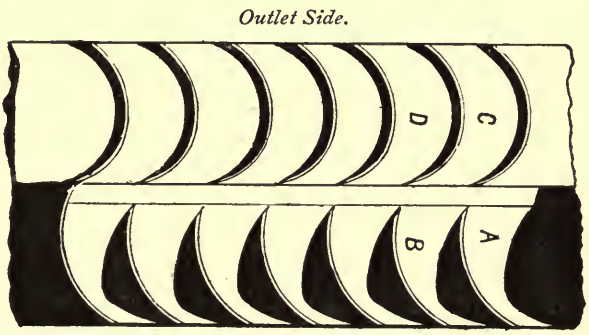


FIG. 162.—Double Face to Face Turbine.

A diagram of this turbine is given in Fig. 163, in which B is the one wheel and C the other, mounted on separate shafts; D is the steam chest and nozzles which converge on the first wheel B. The two wheels are geared by helical cut gearing on to one shaft, the one wheel F being an external toothed wheel, and the other G an internal toothed wheel; with same pitch line they thus both drive in same direction.

This design is the author's, and experience proves it to be of considerable advantage for smaller powers. It will be more fully described later on.

In the ordinary De Laval turbines, if we take, for instance, a 10 horse-power machine, the peripheral speed is 617 feet per second, the revolutions 400 per second, or 24,000 per minute. In my combined wheels, in which the nozzles rotate backwards at same speed as the wheel goes forward, the speed of each wheel is 200 feet per second and 12,000 revolutions per minute. And as the disc stresses are as the square of the speed, they will be reduced to a fourth of what they are in a single wheel, and the skin friction to a fifth.

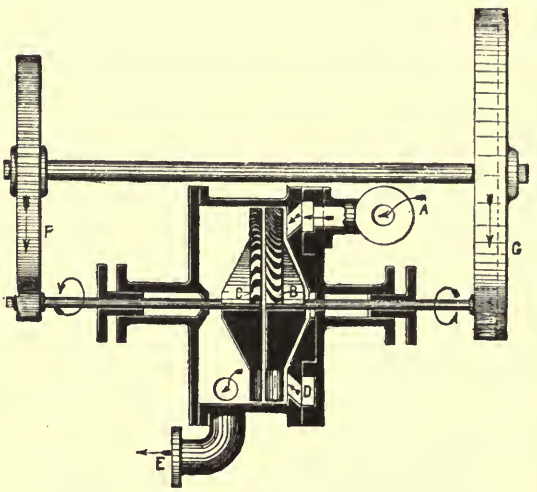


FIG. 163.—Combined Wheels Type of Turbine.

The gearing is objectionable for large sizes. Another direction for experiment lies in the use of long nozzles wound into a scroll on the periphery of a disc, or cut on the face of a disc and closed by a steel disc. The steam being continually deflected exerts a tangential pressure. A rough test of this principle gave fair results, as also did a test with a screw cut on a barrel and gradually increasing in depth towards the outlet. The screw was cut on a brass barrel, 6 inches diameter outside, 5 inches inside. The screw was cut with a square thread $\frac{1}{4}$-inch pitch, the thread being $\frac{1}{16}$ inch thick, and the space between $\frac{3}{16}$ inch; the depth of the thread was $\frac{1}{4}$ inch. After the barrel was threaded it was tapered down in the lathe until the thread at one end was only $\frac{1}{64}$ of an inch deep, so that it gave a long spiral tapering nozzle, tapering 16 to 1 from end to end. A brass sleeve bored to the same taper was slipped over the barrel hot, and shrunk on to it, thus closing the long spiral nozzle. The barrel was mounted on a spindle, one end of which was hollow for steam admission to the narrow end of the spiral. The steam expanding and flowing along the spiral exerted a tangential pressure, which propelled the barrel on its axis. This turbine is shown in Fig. 164. It was not so effective as the scroll.

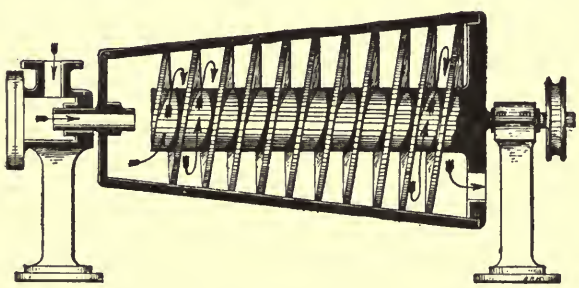


FIG. 164.—Screw Type of Turbine.

Professor Hewitt tried a screw form of turbine, but from the illustrations given of it, of which Fig. 165 is one, the chief direction of pressure would be axial with a long pitch screw working a fixed barrel, whereas it is tangential pressure which is necessary to rotate the shaft.

Mr. Morton tried an experiment on a curved nozzle with steam. He let little glass tubes A, B, C, D, E, and F into the inner and outer curved walls of the tube or nozzle

G, as shown in Fig. 166. The tubes A, B, and C on the greater curved side were U-tubes, and had mercury about half filled up; the tubes D, E, and F on the smaller curve

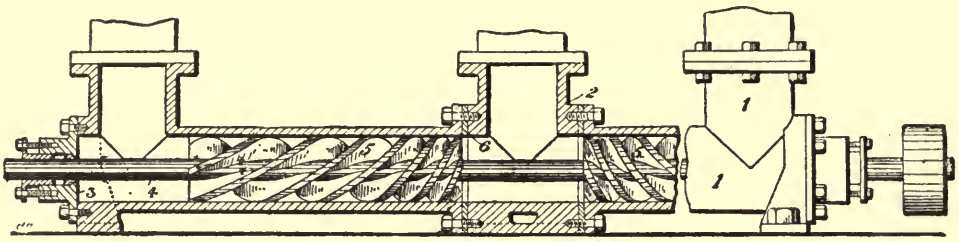


FIG. 165.—Professor Hewitt's Screw Type of Turbine.

dipped into mercury. On blowing steam through the nozzle a considerable pressure was found on the outer curve, and a partial vacuum on the inner curve.

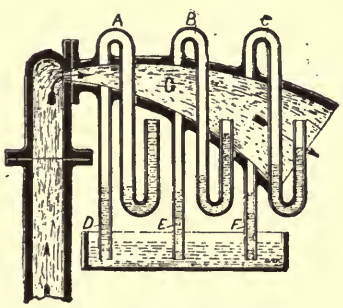


FIG. 166.—Morton's Experiment.

The position of the Hero turbine to-day may be summed up by saying that it is at its best when constructed with nozzles diverging and delivering into another turbine wheel with curved buckets—these two wheels geared to one shaft or driving two dynamos in series with each other, and rotating in opposite directions.

The reaction turbine is one in which the steam passes from the guide blades under pressure, and with some velocity and pressure, so that the driving effort is partly due to unbalanced pressure, as in the Hero machine, and partly due to the impulse from the weight of steam thrown against the curved moving blades.

The purely impulse turbine, the De Laval type, was very thoroughly investigated by Pilbrow, and his results given in his Patent Specification, No. 9658, 1843. He found out that a jet of steam gave a greater impulse to a vane when the vane was some distance from the orifice, and with 60 lbs. pressure; he states that the best distance is $\frac{3}{4}$ inch; and with an orifice of $\frac{3}{8}$ inch diameter he obtained an impulse of 14 lbs. And he calculated correctly that to obtain efficiency the speed in feet per second of the vanes would require to be 1250. Pilbrow allowed the jet to expand and acquire a high velocity before striking the vanes, by shifting the nozzle back until he obtained "the best distance from the orifice; $\frac{3}{4}$ inch from the vanes." It will be seen from this that the steam would have time and space to expand and acquire velocity before entering the wheel. He also shows fixed vanes "to lead away the steam." Their object is not very clear. Pilbrow then put wheels in series, so that the steam entered first one wheel, and then the next, and so on through all of them. We have already referred to his two wheels in series going in opposite directions face to face.

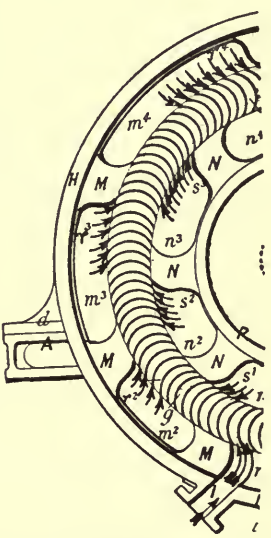


FIG. 167.—Wilson's Turbine.

Robert Wilson, of Greenock, in 1848 patented turbines in which the steam was successively expanded in one wheel, and from wheel to wheel.

A part section of the first form is shown in Fig. 167. The steam enters at I, goes through the wheel to the inside, re-enters at s^1, goes through the wheel to the outside at

m^2, re-enters again at r^2, thence to n^2, and so on, zigzagging out and in, as shown by the arrows, till it finally reaches the exhaust.

In another wheel (Fig. 168) Wilson employed concentric rows of curved vanes alternately fixed and movable, or alternately fixed on two discs or shafts capable of rotating in opposite directions. This turbine has been resurrected with some effect in America recently. The difficulty with this form is the rapid expansion of the steam as it works its way out. Like all multiple wheel turbines, this makes them more suitable for large than smaller powers. In larger sizes the inner radius r can be made a larger fraction of r_1.

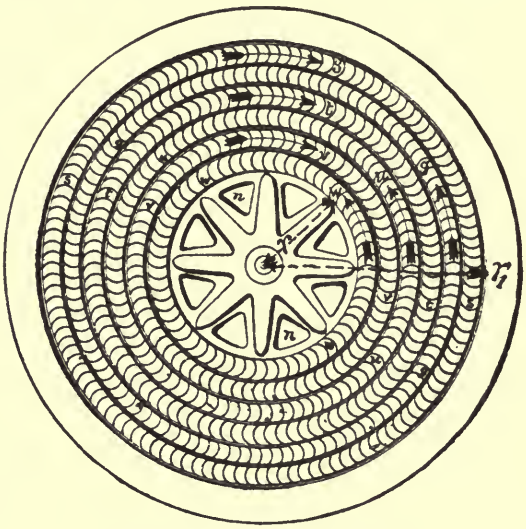


FIG. 168.—Wilson's Turbine.

Then Wilson anticipated the parallel flow turbine, with alternately fixed and movable rows of blades, enlarging their area as he reached the exhaust end.

This is shown in Fig. 169, in which fixed vanes 1, 2, 3 are fastened to the outer casing H, and movable vanes 4, 5, 6 are fastened to a central shaft G. Steam enters at S, and expands through the wheels, each successive wheel being of larger area of wheel blades than the preceding one. There we have the whole of the elements of the best steam turbines now made. All that was required was to design the wheels for high velocity, and the bearings F also for a high velocity.

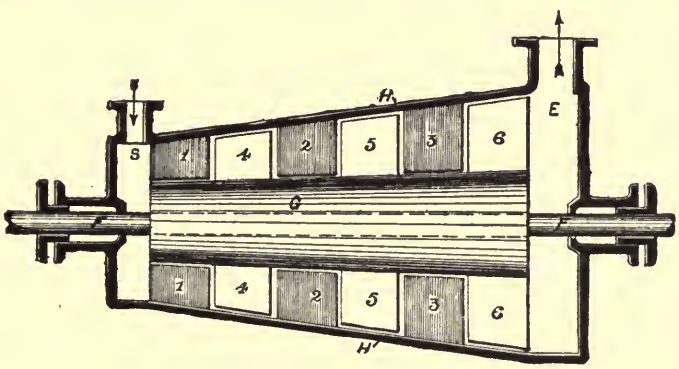


FIG. 169.—Wilson's Parallel Flow Turbine.

It will therefore be seen from this brief review of the pioneer work in steam turbines that most of the principles and designs are very old, and that the pioneers understood the necessities of the matter, especially Pilbrow and his follower Wilson. But they could not make the machinery accurate and fine enough to get the high speeds necessary for practical success.

PRACTICALLY DESIGNED WORKING TURBINES

To Parsons belongs the first place in turbine making commercially and scientifically practical. His patents from the first show that he set out to realise the dreams of his predecessors. First, to design and construct the wheels to work successfully at the necessary high speeds ; and second, to reduce that speed as far as possible without loss of efficiency.

De Laval set out also with the first end in view, and accepted the high speed as unalterable. Both succeeded in making practicable turbines of high efficiency,—the first more especially for large powers, and the second more especially for smaller powers.

There are other practicable turbines more recently introduced, the Rateau and two or three others worth describing.

Matters in turbine progress stood much in the stage at which Pilbrow and Wilson left them until 1884, when Mr. Parsons commenced to wrestle with the difficulties. Greenock men laughed at Wilson's inventions, and his efforts were fruitless to him, but the day came which he had hoped for, when a steamship, the *King Edward*, touched at Greenock propelled by a Parsons' steam turbine, a highly developed specimen of Wilson's type shown in Fig. 169.

It had taken 60 years to develop from Wilson's crude design to the tnrbine on that ship.

Mr. Parsons laboured himself for fourteen or fifteen years before his great improvements were fully recognised by marine engineers. His patents had expired before he began to reap his just rewards. Fortunately, his fundamental one was extended for five years.

This slow progress of inventions in the engineering line is a most remarkable phenomenon. It is difficult to find a reason for it. We as a manufacturing nation go on for generation after generation using and making old things in the old ways as long as it is ever possible to sell them. Some of these things are no doubt past improvement and excellent in their way, but the majority are subjects for improvements. But very slowly and very reluctantly are the improvements adopted. It is wonderful!

The internal combustion engine has the same tale about it. It has in one form or another been in use for the past fifty years, and only now has it been recognised as a prime mover on a large scale and its possibilities recognised. The motor car dates back to the time of James Watt and Murdoch, yet it is only now approaching the time when it will become an important industry. The electric tramways in towns could have been laid down by electrical engineers in quite as complete a system as they are in now in the year 1888, and in quite a practicable manner in 1884, yet it took years and years to introduce them, and they only became fully recognised as the best system of rapid transit in towns about 10 years after their perfection. And then, of course, the authorities began to tumble over each other in their eagerness to secure the advantages of the "novel" system.

The same story could be told of the dynamo and the electric motor. The pioneers did a lot of hard work for nothing from 1874 till 1886. And now every engineering firm with a vertical drilling machine and a turning lathe makes "*the best motors and dynamos in the world*," "our own special design," and so on.

At this moment, when the turbine has been recognised through Mr. Parsons' efforts to be "the steam engine of to-day," and we may say "to-morrow," there is an amusing keenness among engineering firms to enter as rival manufacturers, and by and by each will have a "special design of his own," even although the difference between them is, practically, only in the colour of the paint. The history of the first adoption of the Parsons' steam turbine on mercantile vessels is interestingly and concisely given in Mr. Archibald Denny's remarks on Mr. Parsons' paper on "Marine Steam Turbines," before the Institute of Engineers and Shipbuilders in Scotland, 19th February 1901. As a matter of history it is worth recording:—

"This was not the first time that he had heard Mr. Parsons lecture on this subject: he heard him read a paper to the Institution of Naval Architects about three years ago, and naturally he was impressed with the advantages of the turbine. Curiously enough, his firm had used one for driving the dynamo on board the s.s. *Duchess of Hamilton*, built some years ago, and it worked exceedingly well. When he heard Mr. Parsons read his paper before the Institution of Naval Architects he was fired with the ambition to work with him for the success of the turbine. Several months ago his firm got in touch with Mr. Parsons, and they made up their minds that if possible they would jointly get a turbine vessel built for the mercantile marine. They naturally approached the railway companies in the first instance, but they affected a terrible amount of modesty,

and each company was anxious that somebody else should make the first experiment. So the matter was hung up, and he was beginning to despair of success when Mr. John Williamson came forward and lent them his aid. Mr. Parsons, Mr. Williamson, and his firm, having laid their heads together, resolved to build a mercantile turbine vessel, and he felt it was very gratifying indeed that the Clyde had been favoured in being the pioneer in this enterprise. It was gratifying to think that this was taking place during the era of the Exhibition, when many people would be able to see the advantages of this system."

The attitude of the railway companies referred to is the common one.

To proceed with the turbine construction, we will take the first patent of Mr. Parsons, as that marks an epoch in the art of turbine making.

This invention has reference to motors of the turbine type; that is to say, to motors in which the actuating fluid operates between fixed and moving vanes or blades. When elastic fluids such as gas and steam are used in a motor of this description, it is necessary for economical working that the peripheral speed of the motor should be nearly as great as the velocity of the gas or steam, due to its effluent pressure, a speed which, except with very low pressure, is practically impossible.

According to this invention, to obtain a low effluent or terminal pressure while using a comparatively high initial pressure, a compound motor, or a combination of motors, are so arranged that the same actuating fluid operates therein in a successive manner, undergoing expansion and falling in pressure in each, until it leaves the last at a velocity not greatly above that which is practically attainable by the motor itself, although greatly above that practicable with a motor having oscillating or reciprocating parts. By this arrangement each motor, or successive portion of the compound motor, utilises a portion of the energy of the fluid, and thus, instead of the greater part being wasted as heretofore, it is successively drawn upon until a comparatively high efficiency is obtained.

The motors, or successive portions of the compound motor, may be arranged either upon one common shaft or upon different shafts. In the former case the first will deliver directly into the second, and the second into the third, and so on, the moving vanes of the second (say) rotating between its own fixed vanes, and those of the third, and similarly for the others; the space for the actuating fluid increasing either continuously, or step by step.

This increase may be conveniently gained either by an increased area, or by an increased pitch of the blades, or by an increased area and pitch combined.

The successive motors, or portions of a compound motor, are arranged in such wise as to form an approximately cylindrical figure, the whole being mounted by preference upon one and the same shaft; the first delivering into the second, the second into the third, and so on.

Each motor or portion comprises a set of fixed and a set of moving vanes, the direction of motion of the actuating fluid being generally parallel, or approximately so, to the axis of the combined motors.

Conveniently, each set of moving blades can be formed out of the solid metal on the circumference of a brass or steel disc, the blades extending only about one-third of the breadth of the disc, and the blank portion forming part of the moving cylinder beyond which the blades extend.

Likewise, the fixed blades can be formed by cutting each row internally on a ring which is afterwards cut diametrically into two parts, one to be joined to the top half of the casing, which is divided into two parts by a longitudinal joint, and the other into the lower portion of the casing. When all the parts are put together the fixed portion forms a hollow cylinder with projecting rings of blades, and the moving portion a solid (or hollow) cylinder, also with projecting rings of blades.

To balance the end pressure upon the cylinder, two similar sets of rotary parts are

mounted upon one shaft, one set being placed at each side of the inlet for actuating fluid, in such a way that the entering stream shall divide right and left, and the exhaust take place at both ends. Any end pressure not thus balanced, or due to external causes, can if desired be balanced by pressure of the exhaust fluid acting between the end of the moving cylinder and a collar of smaller diameter than the cylinder. Thus, should the cylinder be displaced endwise in either direction, the exhaust will be checked at that end, and in this way a compensation will be automatically effected.

As the speed of the motor will be necessarily and designedly high, and perfect balancing of the moving parts would not be practicable, the bearings are given a certain very small amount of elasticity or play, combined with a frictional resistance to their motion. Thus the cylinder will be enabled to rotate around its centre of gravity, instead of its geometrical centre, if the two be nearly coincident, and the vibration to which it may be subject will thereby be damped or modified.

The lubrication is effected by forcing lubricant to the parts to be lubricated, and for this purpose a pump can be employed. Conveniently it may be a centrifugal pump, of the type in which the fan is constructed like a screw propeller mounted on the end of the shaft. From this pump the oil will be taken to the bearings as required, with a constant circulation. The oil can also be used as a carrier of heat, to reduce the temperature of the parts liable to grow hot. If the pump be of a kind that will not lift, a suction fan mounted on the motor shaft may be used to raise the oil on the suction side. This fan may also be employed to govern the supply of actuating fluid, by causing variations of pressure, according to the speed at which it is driven, on a diaphragm or piston in connection with the throttle or supply valve. The speed of the motor may be regulated by an adjustable spring acting against this varying pressure, or by the admission of air through a graduated regulating tap into the exhausting side of the fan.

To prevent leakage past the shaft at the end covers of the casing, which, when steam is the actuating fluid, would be inconvenient, annular recesses are formed in the covers around the shaft ends, and place these recesses in communication with a pipe in which a partial vacuum is maintained by suitable means such as a steam jet. Any steam which enters the recesses will thus be drawn away without apparent leakage.

Motors, according to this invention, are applicable to a variety of purposes, and if such an apparatus be driven it becomes a pump, and can be used for actuating a fluid column, or producing pressure in a fluid. Such a fluid pressure producer can be combined with a multiple motor according to this invention, so that the necessary motive power to drive the motor for any required purpose may be obtained from fuel or combustible gases of any kind. For this purpose the pressure producer is employed to force air or combustible gases into a close furnace of any suitable kind such as used for caloric engines, into which furnace there may or may not be introduced other fuel (liquid or solid). From the furnace the products of combustion can be led, in a heated state, to the multiple motor, which they will actuate. Conveniently the pressure producer and multiple motor can be mounted on the same shaft, the former to be driven by the latter; but the inventor does not confine himself to this arrangement of parts.

The blades should be of material that will withstand high temperatures, or water or other fluid may be employed to cool the blades. This may be done by providing, in the cylinders that carry the blades, channels or passages for the circulation of the cooling fluid, which, in the case of the rotary cylinder, may be supplied through a passage or passages in the shaft carrying the cylinders.

It will be observed at once that the inventor tackled the points requiring improvement.

First, to reduce the velocity of steam flow as far as possible, so that the turbine speed would approach in peripheral velocity at least half the velocity of the steam. This he does, as had been done before, by forcing the steam to pass in series through many wheels. The more it is baffled on its way to the exit by deviating its course by wheel

blades, the slower is its progress, and the more of its energy is exhausted by giving it away to the wheels—that is the whole philosophy of the wheels in series.

To balance the steam pressure the turbine was made double ended, steam entering at the middle and balancing. This is still a good plan for pressure turbines; better, in the author's opinion, than the dummy plan later adopted. Then he proceeds to the real problems to be solved. He knows that a perfect balancing of the rotating parts is mechanically impossible; he therefore devises bearings with some elasticity or play in order to allow the "wheels to settle," that is, to rotate round their centre of gravity.

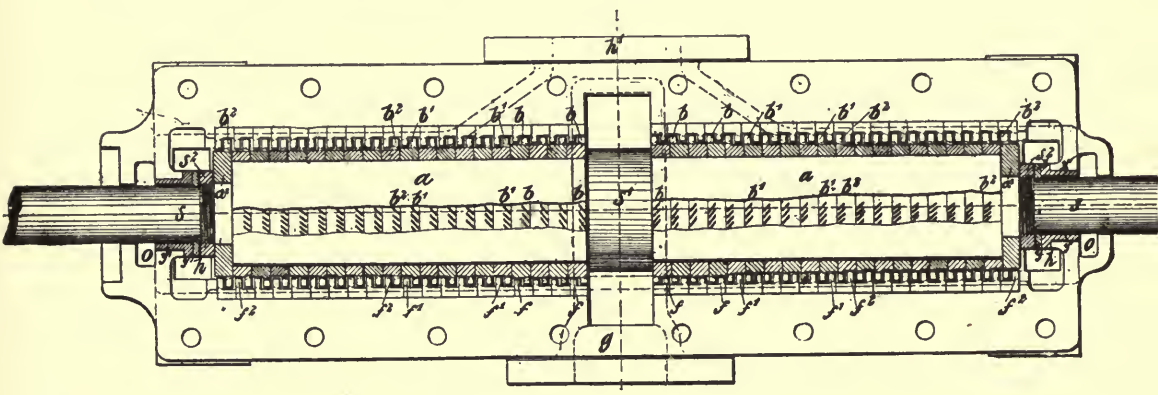


FIG. 170.—Parsons' First Turbine.

The penultimate paragraph in this patent is of interest; it shows that the turbine is reversible, and will act as a pump if driven by power. This, we have seen in Fig. 145, has been accomplished, and then he foreshadows a hot gas turbine in which the air is to be forced into a closed furnace supplied by fuel; the products of combustion are then to be led into the turbine which drives the pump, and so drive the turbine. This scheme has never been put into practice, so far as can be ascertained, and it was dropped out of the specification. It contains, however, the germ of the gas turbine or internal combustion turbine.

Fig. 170 is a longitudinal horizontal cross section of the turbine, with two sets of wheels

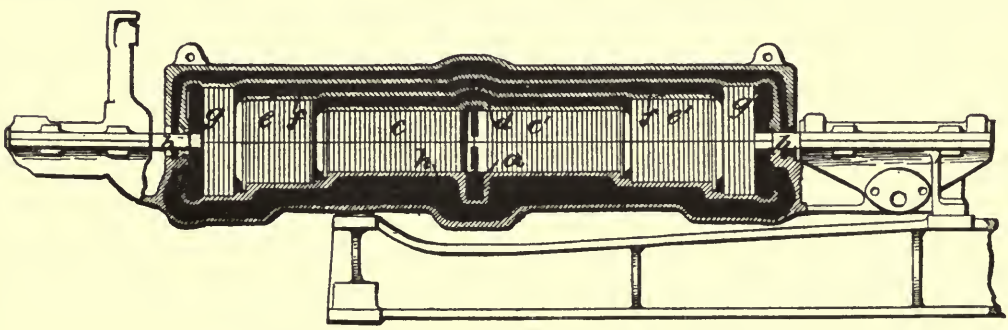


FIG. 171.—Parsons' Turbine, with Expanding Wheels.

right and left of the steam inlet S_1. The b, b_1, b_2 are the movable blades, and the f, f_1, f_2 the fixed blades. These blades were originally cut out of the solid brass rings, and were merely flat blades at an angle of 45° with the axis.

In this form the diameter of the wheels are all alike, and expansion is obtained by making the blades deeper radially as they near the exhaust end.

Later on the plan shown in Fig. 171 was adopted, in which not only are the blades increased in depth radially, but they are also increased in diameter in 3 sections, making it, as it were, triple expansion. And in order to preserve the balance of steam pressure,

steam passages are formed in the casing connecting the annular space between each set and the exhaust ends, so that the pressures are all equalised.

For giving to the bearings a certain very small amount of elasticity or play, combined with a frictional resistance to their motion, in the arrangement of motor now under description, this is done by constructing the bearings of shaft *s* in the manner represented in Fig. 172.

I is a light bush, outside of which are placed metal rings or washers KK^1. The alternate washers K are slightly larger than the washers K^1, so that the alternate washers K fit the casing but not the bush, and the washers K^1 fit the central bush but not the casing. L is a spiral spring, and M is a nut. By these the rings or washers KK^1 are pressed tightly together. Thus the bush is capable of a slight lateral movement, but this movement is resisted and controlled by the rings or washers and their mutual friction. The amount of liberty given to the rings or washers may be, say, $\frac{1}{100}$ part of an inch diametrically, but it may be more or less according to circumstances as may be found desirable in practice. It will be evident that any movement of the central bush in a lateral sense will be opposed by the frictional resistance of the rings or washers. In lieu of, or in conjunction with, rings or washers, such as KK^1 with spiral spring and nut, a steel cage such as illustrated in Fig. 173 may, in some cases, be used to surround the bush *i* for the purpose of maintaining it in place whilst admitting of very slight lateral play or elasticity. This cage *n* as represented is formed with longitudinal slots n^1, so that the intervening parts n^2 will act as springs, pressing tightly against the central projecting part i^1 of the bush, thus resisting any lateral movement principally by elasticity of the cage. To obtain very quiet running, however, it is preferable to use a combination of the two arrangements, constructed by placing a steel cage such as *n* or its equivalent around the washers KK^1 in order to check the communication of vibrations from the rings or washers to the bearing itself.

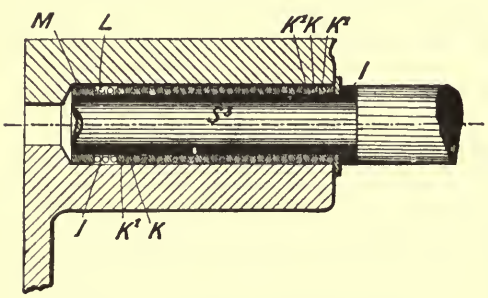


FIG. 172.—Turbine "Ring" Bearing.

In Fig. 170 *yy* are metal washers fitting easily on shaft *s* to prevent leakage of steam past the bushes y^1y^1. *oo* are drain passages from each end of the hollow cylinder. They pass over the top of the motor and meet at its centre. Their purpose is to drain away any of the steam that may escape between the shaft *s* and bushes y^1 fixed in the motor. This is effected by an arrangement shown clearly in Fig. 174. Where the drain passages *oo* meet, there is provided an ejector or steam nozzle *p*, to which steam is admitted from the supply branch. This steam flows into an exhaust pipe *q*, and the partial vacuum thus created causes the steam to flow from the passages *oo* and to pass away through the drain pipe *q*.

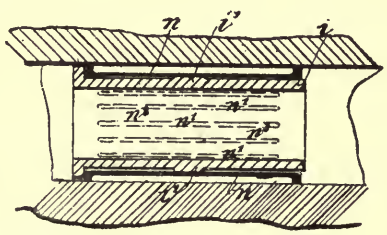


FIG. 173.—Turbine "Cage" Bearing.

His next form of turbine is shown in Fig. 175, a series of radial flow turbines ending in a large disc series of outward flow turbines like Wilson's (Fig. 168). The construction is thus explained :—

It consists of a new arrangement of balance piston designed to minimise leakage ; a method of bolting the rings or discs together on the turbine spindle ; the use of a large disc with blades to expand the steam to a very low pressure when a condenser is adopted, and an application of the principle of the relay to solenoid electrical governors for steam turbines.

Fig. 175 is a sectional elevation of this turbine with balance piston shown thereon, also illustrating the method of bolting the rings or discs together on the turbine spindle, and use of a large disc with blades to expand the steam to a low pressure when a condenser is adopted.

Fig. A shows the balance piston E on a larger scale in part section.

Another part of the invention relates to fastening the discs on the turbine spindle, and for this purpose the spindle A has a collar c^1 at one end, and into this collar c^1 pass long bolts or studs *cc*, which pass through holes in the turbine discs as B, B^1, B^2, B^3, B^4, B^5, and B^6, and through holes in the balance piston E. The discs are thus screwed up and are firmly held on the spindle, together with the balance piston.

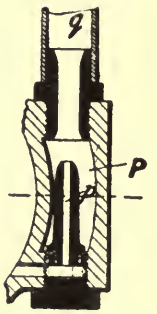
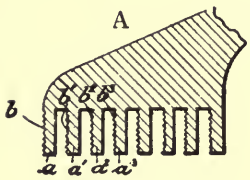


FIG. 174.—Ejector for escaped Steam.

The invention also relates to turbines intended for great expansion and exhausting into a condenser. In these turbines a large disc as B^6 (Fig. 175) is fixed on one or preferably both its faces, and the steam, after passing through a sufficient number of turbine pairs to reduce its pressure to atmosphere, or nearly so, passes by one or both faces of the large disc, and by its impact between fixed and moving blades, gives up its energy, and passes to the condenser, being first reduced in pressure until very little energy is left to be carried to the vacuum. The large disc is arranged as an outward flow radial turbine, and the passage communicating with the condenser surrounds it so that the steam passes on one or both sides of the disc from the turbine through the fixed and moving blades.



This same outward flow disc turbine is shown at K in Fig. 158, already referred to in conjunction with his Hero type of turbine.

Mr. Parsons also patented an inward flow steam turbine, much like a series of vortex wheels as used in Professor James Thomson's

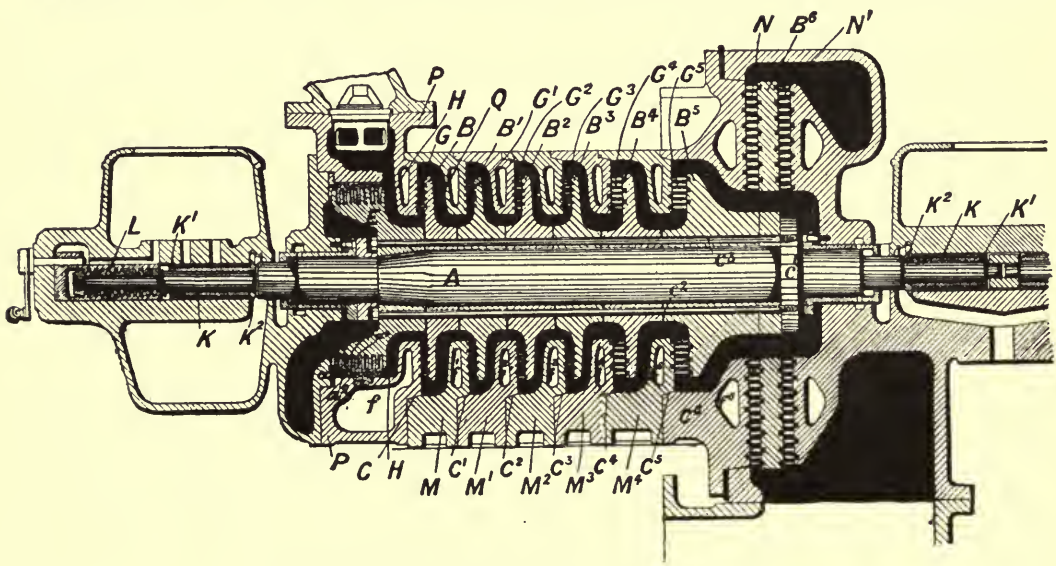


FIG. 175.—Parsons' Radial Flow Compound Turbine.

vortex turbine. It is shown in Fig. 176, a longitudinal section and a part cross section.

In steam turbines of the kind at present in use the steam is caused to perform work by alternate impact upon and reaction from a large number of rotating vanes or blades, and the steam is suitably directed or guided by fixed vanes or blades of convenient shape and disposition. In such motors it is impossible to obtain the maximum

economy in the consumption of steam unless the clearances between the rotating blades and the enclosing conical, disc, or cylindrical surfaces and the fixed blades and rotating surface is very small. This necessitates considerable delicacy in the adjustment and accuracy in the construction of the motors, and renders it difficult to keep them in such condition as to bearing surfaces that the best economy may be obtained in practical work through a long term of years.

As applied to turbines of the inward flow type, a series of inward flow turbine wheels, upon one rotating shaft or spindle and enclosed in one cylinder or case containing the directing vanes or guide blades, in such manner that steam entering the first of the series passes successively through each before being discharged into the atmosphere or condenser. The turbine wheels consist of metallic discs combined with bushes, which are slipped upon the shaft or spindle, and firmly keyed or fixed to it in any other convenient manner. Each disc carries upon a face or faces, or upon an edge or edges, one or more series of blades, of suitable shape and curvature, in which the general direction is that of radiation from the centre towards the circumference. The enclosing case or cylinder, preferably made in halves, carries ring projections and guide blades or vanes, and the ring projections are arranged so as to form a series of chambers in which the turbine wheels rotate. Each wheel chamber may for facility of construction consist

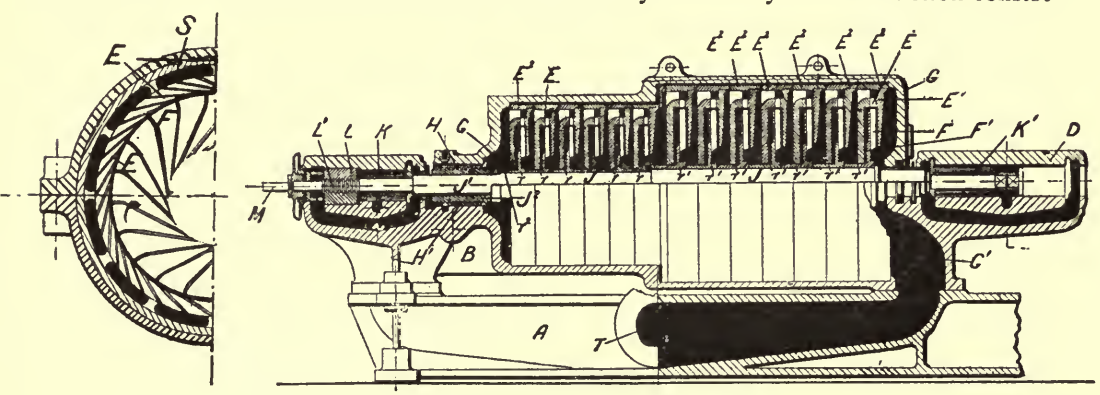


FIG. 176.—Parsons' Turbine with Fixed and Movable Vanes.

of two rings projecting from the interior of the cylinder or case to enclose a wheel in which the steam is caused to pass outwards by the first partition or ring, and then over a space between it and the case to the annular space behind the fixed guide blades or vanes, which are disposed around the circumference of the wheel and direct the flow of steam tangentially upon the blades or vanes of the rotating wheel. After passing the rotating wheel vanes or blades, the steam flows towards the centre and is brought almost to rest, thus giving up the greater part of its energy of motion to the wheel. The steam then flows through an annular space around the rotating bush or spindle, and is directed outwards by the next projecting ring partition to the fixed vanes or guide plates of the next wheel in the series, there to experience a further fall in pressure and a further expansion, and so on until the desired total expansion is accomplished. In order to prevent leakage from any wheel chamber to the next, whereby a portion of steam would be allowed to pass without flowing through the guide blades, the first partition ring of each chamber is placed close to the rotating bush carrying the wheel or close to the spindle itself, and grooves or a series of grooves are cut in the portions of bush or spindle, and ring, opposing each other, which grooves and projections may alternately project into each part without touching or nearly touching laterally or longitudinally. In order to reduce leakage past the faces of the rotating vanes or blades moving near the enclosing partition, a ring is attached to the lower edge of those vanes or blades, or otherwise shroud the blades which project under the fixed partition

ring, and by so baffling the steam diminish leakage. At all surfaces where leakage is likely to occur, which surfaces, of course, cannot be packed or rotated in close contact, alternate recesses and projections, or recesses only, are cut to give the steam a tortuous path and reduce leakage. The fixed guide blades form part of one of the partition rings.

The series of turbine wheels may be mounted on a conical, stepped, or cylindrical spindle, as circumstances may require, and the successive wheels may be of the same diameter or different diameters, generally increasing, as determined by the desired amount of expansion, the number of, and discharge area at, the guide blade orifices ; and the number and configuration of moving blades is also varied to suit various conditions of steam pressure and expansion. The vibration caused by slight want of balance by means of the following contrivance is damped :—

The spindle ends run in a bush fixed into another bush with slight freedom of fit, and this into another, and so on, giving a number of concentric bushes having slight play the one within the other. The ends of these bushes fit into a case with some nicety

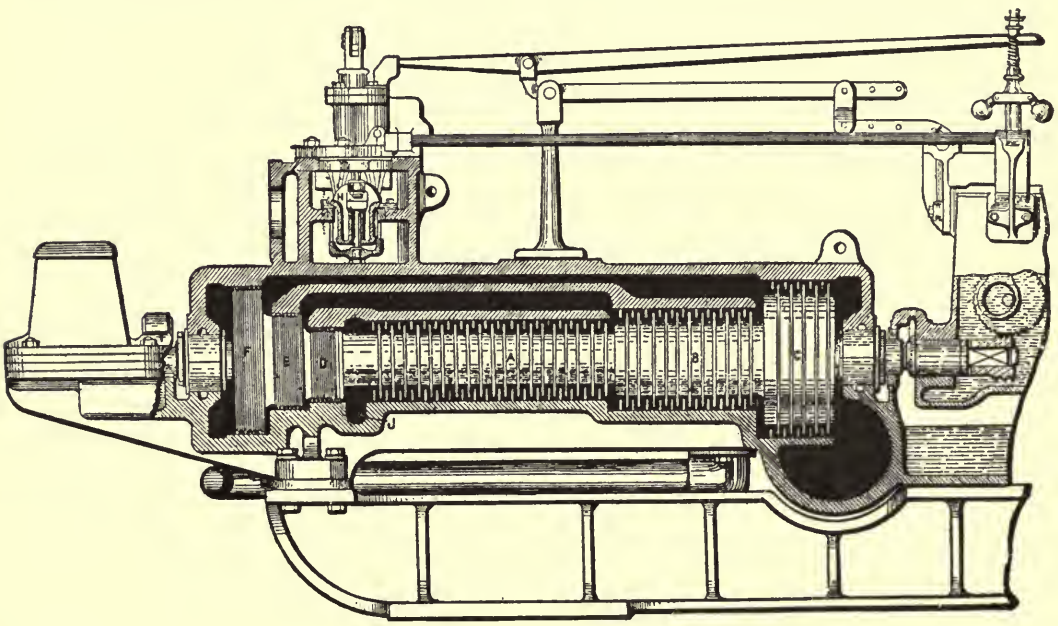


FIG. 177.—Single-Ended Parsons' Turbine.

of end fit, but this is not always necessary, and fill the case with oil, the outer bush being fixed securely. Any vibrating movement is now checked or damped by the forcing out of oil between the bushes and the ends and circumferentially around the bushes, so that although a slight movement is possible, yet it is resisted in whatever direction it may tend to move. Spring supported bearings may be used, and hydraulic resistance of other kinds may be interposed such as that due to a small cylinder or ram. The bush in which the spindle runs is packed with segments of a thin cylinder or tube, which segments may be of greater or less curvature than the outer surface of the bush, and so cause bearing of the segment ends upon the exterior of the bush or the interior of the bored surface holding it. This arrangement gives a spring or elastic movement, and when oil or other liquid is present the hydraulic resistance already hereinbefore described is also brought into play.

In order to prevent leakage of the steam past the spindle, a series of grooves or projecting collars are fitted upon the spindle fitting into or working easily in a similar series of grooves in an end bush, and by using a sufficient number leakage may be

reduced to any desired extent. Instead of this, metallic packing rings may be used. When the steam turbine is arranged so that there is end thrust in any direction, and to obviate end wear for an inevitable small end thrust, that end thrust is taken in a thrust block of construction similar to those used in marine engines; but in order to secure that each thrust collar shall take its share of the pressure, and so produce a uniform slight pressure over a considerable surface, which uniformity is necessary to run cool at high velocities, the collars or flanges have a slight elastic movement, bringing them to bear with a predetermined pressure upon their respective collars on the turbine spindle. Thus they distribute the pressure upon the surfaces accurately, and make certain that no one surface is subject to undue pressure.

A standard type was at last arrived at, as shown in Fig. 177. This turbine is single ended, and balanced by dummy pistons instead of by another turbine.

It was long known that a piston well fitted, but yet smaller in bore than a cylinder in which it can move freely, offers a considerable resistance to the leakage of fluids; and if the piston is long and grooved with many grooves the resistance to leakage is very great. An air pump was used at one time largely working on this principle; it is shown partly in Fig. 178, which shows the barrel and grooved piston. The piston does not touch, the barrel being $\frac{1}{1000}$ of an inch clear; it requires no oiling, and has no friction. Arc lamp dash pots are made on the same principle. In later steam turbines this principle has also been adopted instead of packing. Mr. Parsons carries it further when the piston is to rotate instead of reciprocate; he grooves the cylinder to fit the piston, thus adding still further to the resistance to leakage. The three dummy pistons are thus fitted in the turbine case, and very little leakage occurs, while they balance the steam pressures.

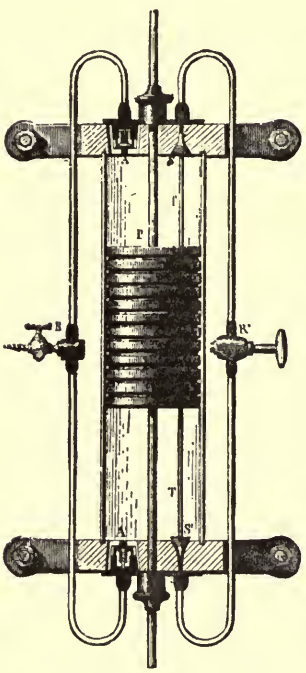


Fig. 178.—Deluel's Piston.

This system of packing steam tight will be again referred to under Cylinder and Piston Engines. With modern machine tools it is possible to fix piston rods and pistons without the old-fashioned packings, spring rings, and other nostrums. Deluel's old air pump contained this valuable principle in its construction, hence it is worthy of reference again to it. With the construction of this type of turbine Mr. Parsons arrived at a permanent design for land engines.

In vessels of the mercantile marine of moderate fast speed it is of more importance to obtain economy in coal consumption than to reduce the weight of the engines and condensers to their lowest limits, as is usually done in torpedo-boat destroyers, where the boilers are extremely light and heavily pressed, and the highest possible speed is the first consideration.

For the mercantile marine, therefore, it becomes desirable to design the turbines for the greatest possible economy in steam, consequently the ratio of expansion extends over nearly the whole range between the boiler pressure and that in the condenser; the condensers are also made of ample size so as to maintain a good vacuum, and an efficient feed-heating arrangement is provided to warm and heat the feed.

The marine steam turbine will be found to be superior or at least equal in economy of coal to the reciprocating engine when placed in fast vessels of the mercantile marine; but it may be asked, What will be the economy of the turbines when, as in the case of yachts and almost all war vessels, much steaming is done at from one-eighth to one-tenth power, the full power being only occasionally used? The answer is a simple one. At

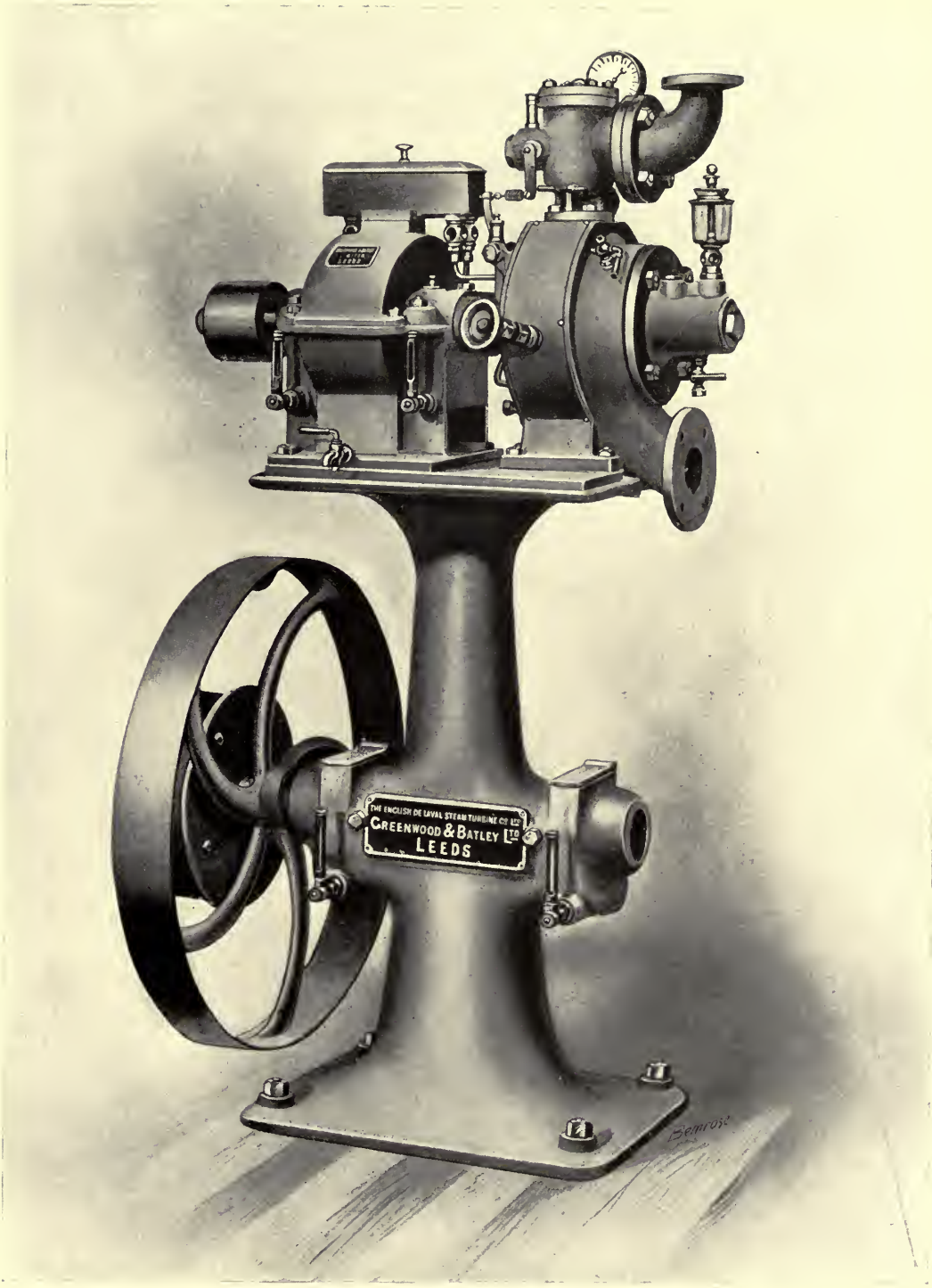


Plate VI.—Small De Laval Turbine Motor.

cruising speeds the revolutions of the turbines fall well within the limits of speed of small reciprocating engines, and such small engines are then directly coupled to the main turbines, and work in conjunction with them, these small triple-expansion reciprocating engines taking the steam directly from the boilers and expanding it down to about atmospheric pressure; it then passes to the high pressure turbine, and thence through the low pressure turbines to the condensers.

It should, however, be added that the turbines alone have their full measure of economy from half to full power, and even at one-quarter full power the economy is good. It is only when the cruising speed falls to nearly one-half the full speed, and the horse-power one-eighth of full power, that the economy of the turbines requires some assistance, and such additions are only necessary in vessels such as war vessels and some yachts, where much running is done at these very low speeds. They are quite unnecessary in passenger vessels and liners.

The arrangement of turbine machinery for an Atlantic liner of 20,000 to 30,000 indicated horse-power presents no features of novelty over the preceding designs, but its simplicity of construction, as compared with the present usual reciprocating engine, is more apparent than in smaller vessels.

The adaptation of steam turbines to marine propulsion required two efforts. The screw propellers in use are all slow speed, the turbines of high speed; hence propellers have had to be designed to bring up their speed to the lowest possible speed of the turbines, and the turbines had to be designed to drive at the highest speed permissible by the screw propeller.

The next improvements relate to marine engines. The steam turbine is not an engine in which the motion can be reversed readily, and in locomotive work and marine propulsion reversing is a necessity, while in stationary engines it is seldom necessary except in the case of mining winding engines. Pilbrow proposed to use turbines on locomotives, but as yet no locomotive has been successfully worked by them. And it is curious also that the internal combustion engines are not yet made reversing. In marine steam turbines Mr. Parsons has met the difficulty very well by employing separate reversing turbines of smaller power than the main turbines. Mr. Parsons' views on the marine steam turbine were given in his paper already referred to.

The most important field for the steam turbine is undoubtedly in the propulsion of ships. The large and increasing amount of horse-power, and the greater size and speed of the modern engines, tend towards some form which shall be light, capable of perfect balancing, and economical in steam. The marine engine of the piston type does not entirely fulfil all these requirements.

Though for obvious reasons up till the present time turbines have only been fitted in vessels designed for phenomenal speeds, yet it must not on this account be assumed that they are only applicable to such vessels. The two conditions of suitability are that the vessel shall have a moderately fast speed and be of moderately large size. For slow vessels of moderate and small size the conditions for turbine machinery are not at the present time so advantageous.

This will appear clear when we consider that the turbine machinery is actuated by the momentum of the steam, and that the rows of blades must be sufficiently numerous, and must move at a sufficiently high velocity to secure a good efficiency from the steam. The class of vessels that are most suitable for the application of turbine machinery are the following: Pleasure steamers, passenger and cross-channel steamers, liners (including Atlantic liners of the largest size), also all fast war vessels, such as torpedo-boats, destroyers, cruisers of all sizes, protected cruisers, and all battleships of the usual speeds.

Now, considering that turbines are not reversible in motion, and that they are best driven at high velocities far beyond the permissible speed of a screw propeller, it seems to my mind that Mr. Parsons might consider an intermediate transmission gearing between

the turbine and the propeller. The introduction of an auxiliary reciprocating engine as proposed, along with a reversing pair of turbines and the special screw propellers, somewhat detracts from the advantages of the simplicity of turbines; and if all that extra machinery could be replaced by some transmission of power system flexible enough for speed variations and reversing, some considerable advantages might accrue. In my own view, there are two systems which might be tried: first, an electric transmission. Let the turbines run dynamos of small size at high speed, and let the current operate motors on the propeller shaft at moderate speeds. With modern electric machinery this should present no difficulties; even in large ships the control would be perfect.

The other system is a hydraulic one, in which the turbines would pump water to a pressure to work by water jet propellers. Properly designed, this would also be feasible, at any rate, for small craft. In this way the turbine speed could be adapted to the best speed for economy.

The marine turbines are much the same as the others, only they are of much larger diameters of wheel, and, as they drive usually on three shafts, one or more of them has a reversing turbine and a main turbine on one shaft and in one casing. Probably the application of the steam turbine to marine propulsion has been the most important step, and we shall therefore fully examine the various inventions of Mr. Parsons for this purpose. The applications of the turbine to screw propellers and their arrangement in the ship will be treated in the chapter on "Marine Engineering."

In Patent No. 14,476, 1899, we have several marine steam turbines specified, principally with a view to combining ahead with astern turbines. As no turbine has been designed to drive in either direction, it is necessary to combine two turbines to do the double movement, and this has been accomplished by Mr. Parsons in several ways. Thus from the specification, the new invention comprises a combination of main steam turbine with reversing steam turbine in which the main turbine and reversing turbine are enclosed within a casing formed in one piece, or within a casing formed of several pieces bolted together, the exhaust ends of both main and reversing turbines discharging directly into the condenser, or into one passage leading to the condenser—a combination in which the reversing turbine is telescoped within the main turbine in order to reduce the longitudinal space occupied, and the reversing turbine case preferably then revolves while the centre part is fixed; and a combination in which the reversing turbine is placed at the steam end of the main turbine, and the exhaust from the reversing turbine passes through the interior of the main turbine spindle or drum. In all these modifications the reversing turbine is one of ordinary types, and it is run in the vacuum of the condenser while the main turbine is in operation.

Referring to the illustrations, Fig. 179 is a sectional elevation of a main and reversing turbine mounted on the same spindle, and arranged and constructed according to one modification. Turbines end to end are used in this case.

Fig. 180 is a sectional elevation of a main and a reversing turbine mounted on the same spindle, and arranged and constructed according to another modification.

Referring in the first place to Fig. 179, *a* is the main turbine enclosed chiefly in the casing *c*, and *b* the reversing turbine enclosed in the inner part *l* of the casing *d*, the casings *c* and *d* being bolted together at *e*. The main turbine *a* is of the well-known parallel flow type, and is mounted on the spindle *f*, which also carries the reversing turbine *b*, which has its low pressure end *h* turned towards the low pressure end *g* of the main turbine. Both low pressure ends open into the low pressure casing *d*, and discharge by the same passage *k* to the condenser; or the low pressure casing *d* may be made part of the condenser without special passages. The steam supply for the main turbine enters by the passage *n*; the steam supply to the reversing turbine is carried through the low pressure casing *d* by a separate steam pipe *m*. By this arrangement a powerful reversing turbine which rotates in the condenser vacuum

when the main engine is operating the spindle *f* in the usual direction, *e.g.*, for propelling a vessel ahead, is provided.

Instead of arranging the vacuum ends together, the reversing motor may be turned round, placing the high pressure end near the vacuum end of the main engine, while still maintaining the separate turbine casing *l* within the low pressure casing *d* and

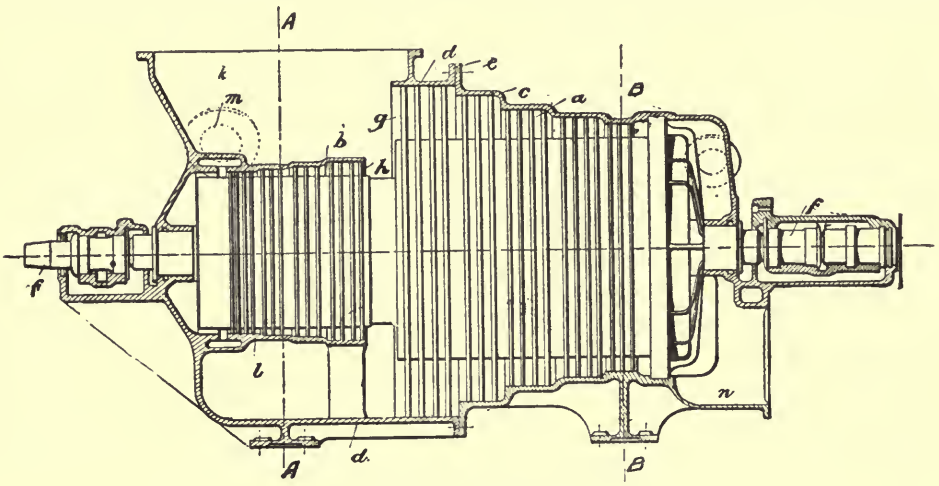


FIG. 179.—Parsons' Marine Turbine.

discharging from the reversing turbine into the low pressure casing as before, and also running the reversing turbine in the condenser vacuum as before mentioned.

Referring now to Fig. 180, which illustrates the second modification, the reversing turbine *b* is here telescoped within the main turbine *a* in order to economise longitudinal space. The spindle *f* carries the drums *o* and 7, which support the rotating blades of the main and reversing turbines respectively. The fixed blades of the main turbine are attached as before to the outside casings *c* and *d*, and those of

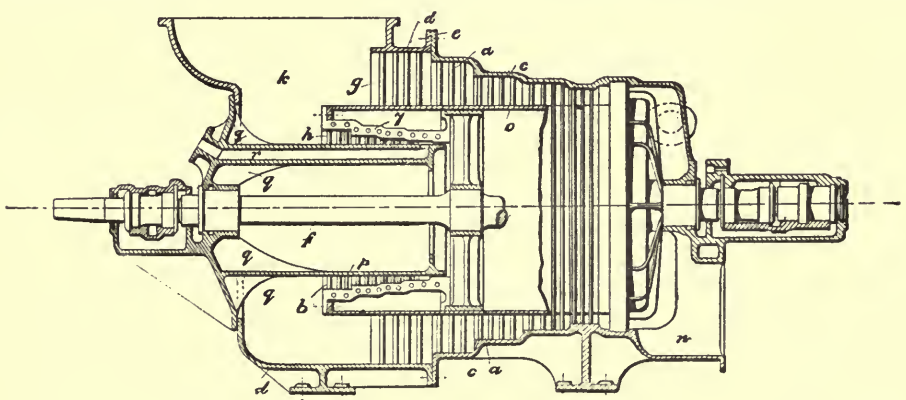


FIG. 180.—Parsons' Telescoped Marine Turbine.

the reversing turbine are fixed on the outside of the inner cylinder *p*, which is preferably supported from the exhaust casing *d* by projecting arms or webs *q*. Steam is supplied to the reversing turbine by a pipe or pipes *r* inside, or forming part of the inner cylinder *p*. These steam pipes may serve for fixing the cylinder *p* to the end of the casing *d*. Instead of this arrangement the steam may be conveyed to the reversing turbine through the spindle *f* from the passage *n*, which supplies steam to the main turbine as in the previous modification.

Another form of turbine is shown in Fig. 181. To prevent leakage from the steam end of the main turbine to the steam end of the reversing turbine, or *vice versâ*, at points 2 and 3 baffle grooves and rings are formed on the casing *c* alternating with a series of grooves and rings on the ring *v* in order to check the flow of steam, and holes are provided on the ring *v* to allow any leak of steam to escape into the condenser.

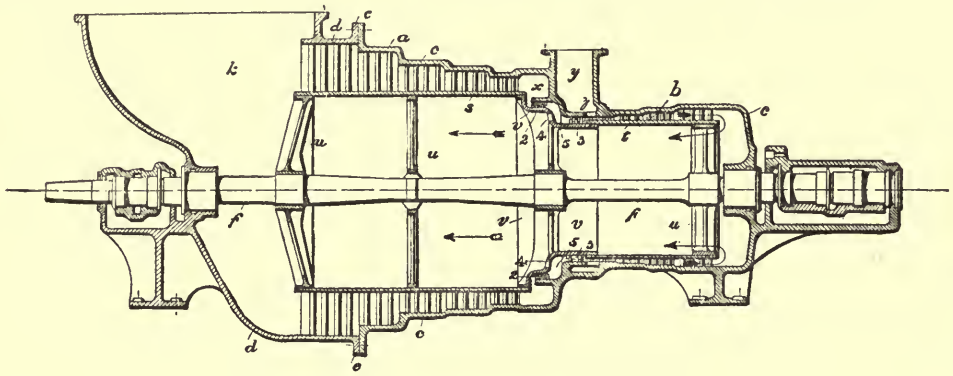


FIG. 181.—Double-Ended Marine Turbine.

In all the modifications the reverse turbine runs idly in the vacuum when not in use.

The telescoping arrangement can, of course, be used to make a compact turbine for land purposes.

The next improvement, made in conjunction with Mr. Wass, relates to connecting turbines directly to condensers. They had found by experiment that a very large

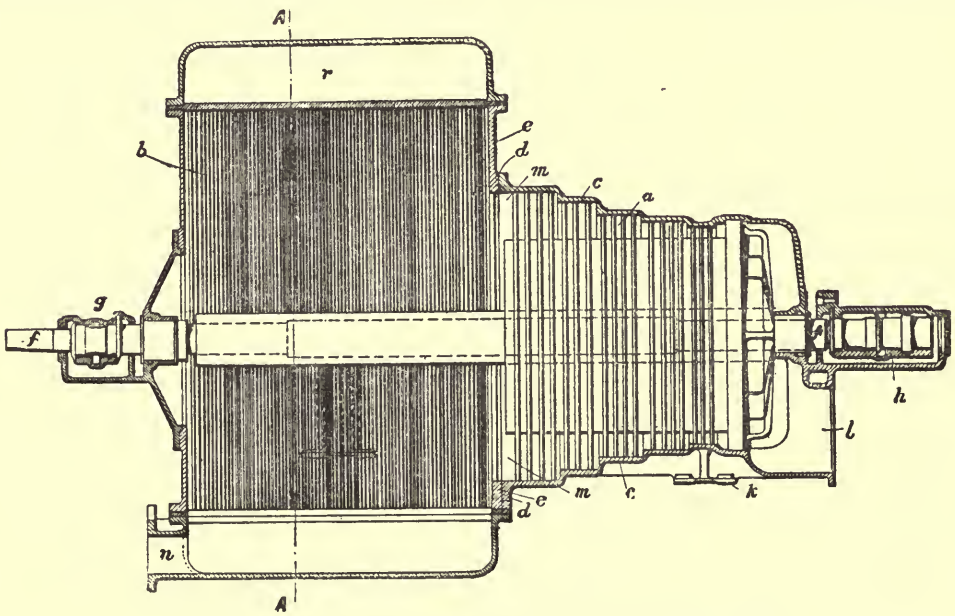


FIG. 182.—Parsons' Combined Turbine and Condenser. Section.

proportion of the total energy of steam can be obtained with a steam turbine by expanding to a very low pressure—such a pressure, for example, as one-thirtieth of an atmosphere. By this a turbine is found to give considerably increased efficiency. This increase of efficiency, however, is easily lost by introducing a slight back pressure, and it is difficult on board ship, or in confined spaces, to obtain room for an exhaust pipe of sufficiently large dimensions to prevent injurious throttling of the exhaust.

The invention consists in entirely dispensing with connecting pipes between the low pressure turbine and the condenser, and, in fact, making the low pressure turbine and the condenser in one part, or as parts of the structure which may be built up of several parts, the condenser being formed of one or more groups of tubes. The steam thus leaves the low pressure turbine blades and passes directly to the surface condenser part of the turbine cylinder without any intermediate connecting pipes whatever.

Referring now to the illustrations—

Fig. 182 is a sectional elevation of one modification of the invention, in which a low pressure steam turbine is bolted to a surface condenser.

Fig. 183 is a plan of the same.

a is the turbine, and *b* the condenser. The casing *c* of the turbine is bolted at *d* to the end *e* of the condenser. The shaft *f* passes through the turbine and condenser, and is supported in bearings at *g* and *h*. *k*, *k* are the feet or supports of the turbine and condenser.

The steam enters the turbine at *l* and passes through the turbine doing work and leaving the last vanes, passes at *m* into the condenser, which is of a usual construction. The outlet from the condenser to the air pump is shown at *n*. The circulating water

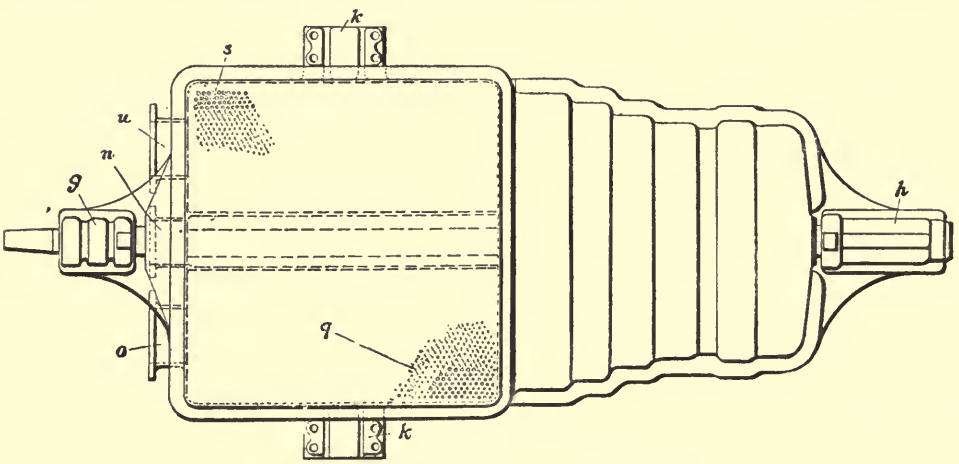


Fig. 183.—Plan of Turbine and Condenser.

enters the condenser at *r*, passing into the end space *p*; it then passes up through the tubes *q* to the top compartment *r*, whence it descends through the tubes *s* to the compartment *t*, and leaves the condenser at *u*.

The low pressure blades of the turbine thus revolve at but a short distance from the surface condensing tubes, and no resistance is experienced in passing the enormous volume of exhaust steam at a low pressure from the turbine blades.

By this arrangement the steam expands down to a very low pressure indeed, without losing the effect of expansion by back pressure or undue resistance between the turbine and the condenser.

Mr. Parsons had five years previously patented some claims for this utilisation of very low pressures in his turbine. We will only select three figures from the Specification, No. 367, 1894. This invention relates to a combination of a reciprocating steam engine with a steam turbine, and its object is to increase the power obtainable by the expansion of steam beyond the limits possible in reciprocating engines. He discovered that pressures much too low to be utilisable in the low pressure cylinder of a reciprocating condensing engine may be made to develop very considerable power in a steam turbine of simple construction specially designed for low pressures. The invention consists in combining together an expansive reciprocating steam engine

and a low pressure steam turbine in such a manner that when the high pressure steam has acted in the reciprocating engine, and has been expanded as much as possible in the low pressure cylinder, it passes to the low pressure steam turbine, and thence, after further expansion, to the condenser.

The work done by the steam in the cylinders of the reciprocating engine is thus supplemented by the work done by the steam turbine, and so the power obtained from a given weight of steam is largely increased. It will thus be seen that the turbine forms the last element in the series of apparatus for utilising the energy of the steam on its way from the boiler to the condenser, thus substantially increasing the efficiency of the engine.

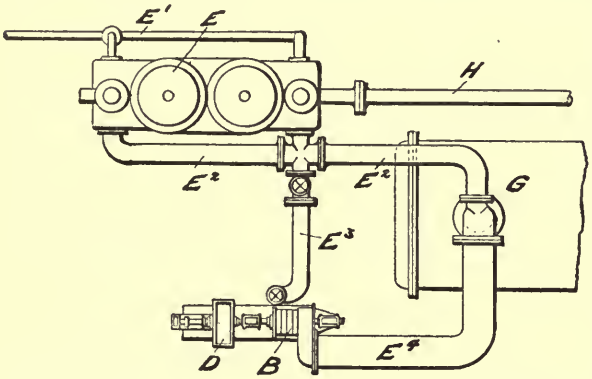


FIG. 184.—Parsons' Combined Engine and Turbine. Plan.

Fig. 184 shows in plan a reciprocating compound condensing marine engine E combined with the low pressure turbine B; the turbine being utilised for any power purposes in the ship.

The turbine B may drive a dynamo D and produce electricity to light a mill, or be used for working centrifugal pumps or fans for forced blast, urging the fires, or any other purpose.

A section of the turbine B is shown at Fig. 185 coupled directly to a dynamo. B is the low pressure turbine, having a large disc B^1 carrying rings of blades B^3 on both sides of it; these rings of blades alternate with rings of fixed blades in the turbine case. The exhaust steam from the condensing engine enters the turbine by the passage B^2 as indicated by the arrow; it passes radially outwards through the rings of blades on both sides of the disc B^1, as shown by the arrows 1, 2, 3, 4. The disc is perforated at the centre to allow the steam to pass through. From the disc the steam passes to the condenser B^4, as shown by the arrow 5. D is a dynamo coupled directly to the turbine spindle.

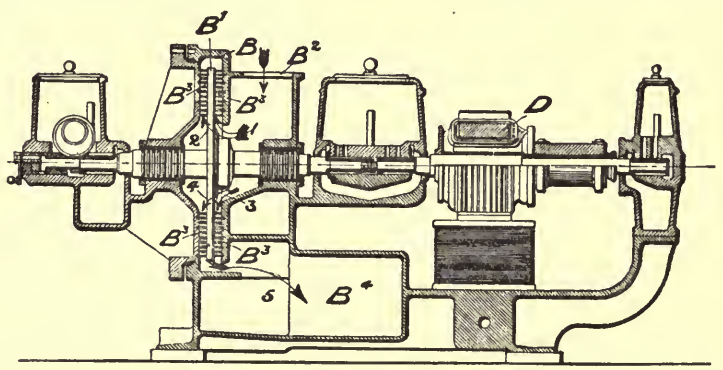


FIG. 185.—Section of Exhaust Steam Turbine. Radial.

Fig. 186 shows the parallel flow turbine adapted for low pressures.

In Fig. 184 the reciprocating compound condensing engine E is shown in plan as driving the propeller shaft H. It is supplied with high pressure steam by the pipe E^1, and the exhaust steam passes by the pipes E^2 to the condenser G, a pipe E^3 from the exhaust pipe E^2 supplies low pressure steam to the low pressure turbine B, and the exhaust from the turbine passes by way of the pipe E^4 to the condenser G. In this case the turbine B is utilised for the electric lighting of the ship, and it actuates the dynamo D coupled directly to its spindle.

Instead of, or in conjunction with, the dynamo, the turbine may drive fans for

ventilating or forced draught purposes, or pumps for circulating water or discharging water from the ship.

Stop valves are provided to enable exhaust steam to be supplied to the turbine, so

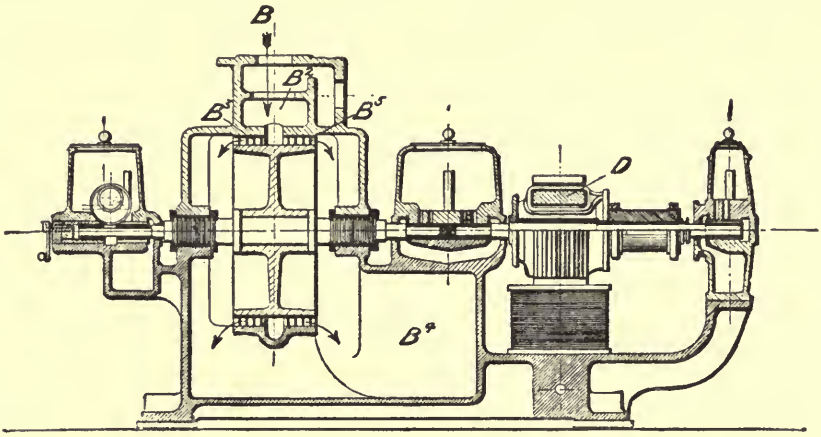


FIG. 186.—Section of Exhaust Steam Parallel Turbine.

that the whole of the exhaust steam from the main engines may be passed through the turbine if required, or part passed to the turbine and part to the condenser direct.

These simple forms of turbines might, and no doubt will, be used in many trades where boiling is a process consuming much fuel, the waste steam from which would give considerable power if passed through a turbine into a condenser.

And it is a moot question whether the best way to compound an engine with high pressure reciprocating pistons would be to connect it with a turbine.

In any case, it is important to bear in mind that a turbine, working with steam at or below atmospheric pressure into a good vacuum, recovers a great deal of the heat energy.

By interposing a simple turbine in the exhaust steam pipe between the engine and condensers in compound or simple reciprocating engines it may pay to obtain electric light in that way.

BLADES OR VANES OF PARSONS' TURBINES

These at first were formed by milling teeth on brass discs, forming flat blades with a passage between of 45° angle with the shaft.

These, however, were not highly efficient, as the steam in being deflected was partly thrown against the wrong side of the passages, thus retarding the movement.

By properly curving the blades and shaping the passages the steam in passing presses wholly on one side, so that at an early date Messrs. Parsons adopted the form of blades shown in the illustrations.

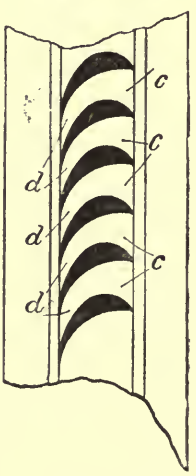


FIG. 187.

Fig. 187 is a view of the blades seen end on as arranged on a turbine wheel.

The invention consists in forming the blades upon the back, that is, on the convex side as compared with the concave side, with a thickening or projection as at *a* (Fig. 187), the said thickening or projection being nearer to the edge where the steam enters than where it discharges. The projection *a* tapers off to both edges, but more

gradually to the discharge edge than the admission edge. This is clearly shown at Fig. 187, where *c* are the admission edges and *d* the discharge edges.

By this modification in the shape of these curved blades a passage between the blades is produced in which the minimum of loss of energy occurs presumably by the reduction of eddy resistance, but possibly in other ways. In any case, whatever be the correct explanation, an improvement of about at least 10 per cent. was found in steam consumption by altering one of the steam turbines in this manner.

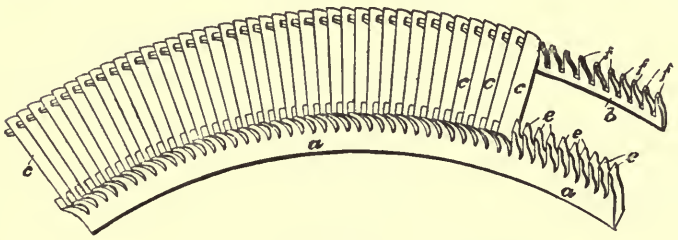


Fig. 188.

In one case, for example, a moderate amount of thickening, such as shown in Fig. 187, at the left-hand side, gives a gain of 12 per cent.; and that a greater thickening, such as shown at the right-hand side, gives a better result still.

These blades are formed into circles or segments of circles by a very ingenious and simple tool patented by Mr. Parsons and others.

As applied to the parallel flow turbine, suitable strips of ductile metal, preferably brass, are bent into a circle or sector of a circle. On one edge of the strip teeth of a special shape are cut by mechanism. The form of the teeth is such that when the blade sections are laid in the grooves and the teeth turned over upon them the teeth and blades fit each other closely and form a secure fastening or mechanical joint. This joint is sometimes made more secure by notching or perforating the blades before insertion, so as to interlock with the strip. One or more rings or shroudings may be put upon the blades. The blade sector so formed is inserted and caulked into grooves in the turbine drums and cylinders, or is otherwise secured to them.

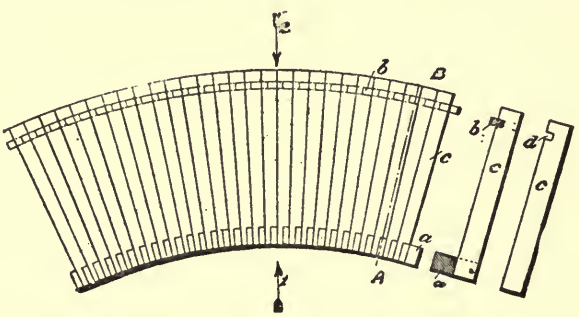


Fig. 189.

Referring to Figs. 188 to 190, *a* is a heavy shroud having teeth cut into one side, leaving, however, sufficient breadth of metal uncut to provide a strong strip to serve as base for the blades; *b* is a light shroud which has also cut teeth, but the uncut breadth is less than in *a*. *c*, *c*, *c*, etc., are blades which are laid in the spaces between the teeth of the two shrouds, and each tooth is closed over on the blade held by it. On the right of Fig. 188 teeth are shown in the position occupied before closing over; the teeth of the heavy shroud are lettered *e*, *e*, and those of the light shroud *f*, *f*; the blades *c*, *c* are notched at *d* (Fig. 189), and the light shroud engages with the notches as shown.

The bases of the blades may be also notched or drilled to increase the grip between the teeth of the heavy shroud. The teeth *e*, *e*, *f*, *f* (Fig. 188) are formed with a discontinuous outline on the back, so that when bent over and closed on the top of the blade the convex outline shall then be a continuous curve, which corresponds exactly with the concave surface of the blade, and fits that surface. By this means the teeth bear solidly all over the ends of the blades, and form a strong continuous sector or ring as shown.

We may cut the notches between the teeth to such a depth that the teeth when bent over will project above the blades, and the surplus metal may be subsequently

turned off, leaving the shroud parallel, so as to exactly fill the grooves in the drums or cylinders.

Both the grooves in the drums and cylinders, and also the shrouds, may be dovetailed slightly for better security, and when the blade rings have been placed in position the teeth *e* are separately expanded by caulking, so as to fill the grooves and hold the blades tightly. Sometimes they prefer to place a strip of soft metal alongside the shroud in the groove, and caulk the strip to secure tightness of the whole structure.

In this same specification plans for casting the blades into shrouds are described and illustrated, but the mechanical method is seemingly much preferable.

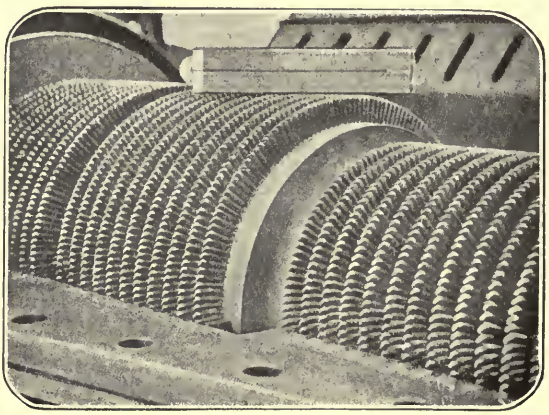


FIG. 190.—Parsons' Wheels.

In Fig. 190, given in Mr. Parsons' Glasgow paper, is shown the complete construction of the revolving wheels in the three sets of wheels in one casing, the top half of which has been removed, showing the curved blades; and Fig. 191, from the same paper, shows one of the sets of engines for H.M.S. *Destroyer*, in course of erection; and Fig. 192 the complete steam turbine now made in large sizes.

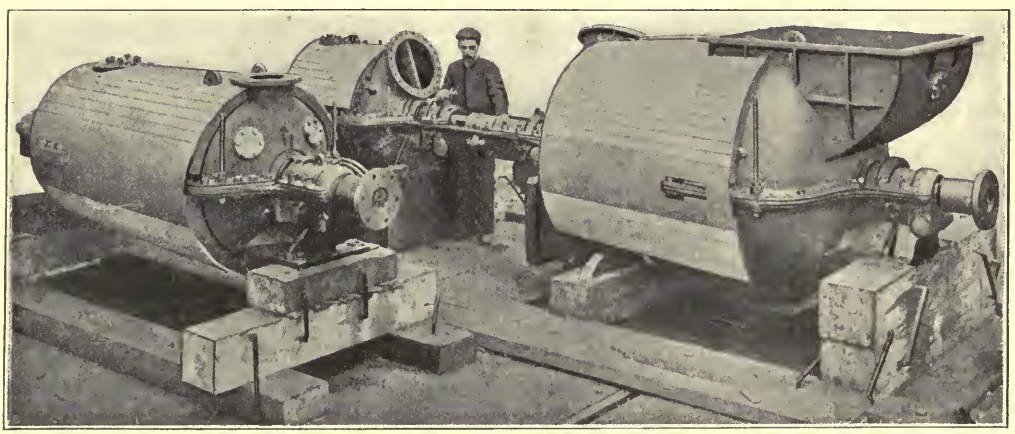


FIG. 191.—Parsons' Marine Turbines.

DE LAVAL STEAM TURBINES

We now pass on to the other pioneer in steam turbines who struck out in original lines to secure efficiency. In the early 'eighties De Laval had invented cream separators working on the principle of centrifugal force. He therefore, in order to get the highest practicable value for this force, was compelled to drive the separating vessels at enormous speeds. He was thus brought into contact with the difficulties due to high speed driving, and became thus practically acquainted with them; and, as a direct way of obtaining a high speed from steam power, he used a Hero turbine directly coupled to his cream separator shaft; and from these beginnings he started out on the improvement

of steam turbines, culminating in the successful production of a wheel and nozzles constructed and operated on thoroughly scientific lines and with novelty therein.

With the true modesty and dignity of the genuine inventor, the first important patent is contained in the following few lines. (Patent 7143, 1889) :—

"My invention relates to an improvement in turbines which are set in motion by means of a current of steam; and the object of the improvement is to increase, by complete expansion, the velocity of the steam current, thus producing the relatively largest quantity of *vis viva* of the steam.

I attain this object by the construction of the steam supply pipe in such a manner that the cross sections of the same are slowly increased near to the turbine wheel and in the direction of the latter. The ratio of increasing the cross sections is due to the proportion and distance between the smallest section and the largest one, in such a manner that in the steam passage between these two sections a permanent current of steam is produced under isoëntropical expansion.

Referring to the illustration, Fig. 193, a front view and side elevation, both partly in section, which shows the mouthpiece of a steam supply M, constructed as above described, in combination with a turbine wheel A, *b* is the smallest and *c* the largest cross section. Between both these sections the steam expands from the pressure 0.577 P_0 (P_0 = boiler pressure) to the pressure of the receiver (= P_2).

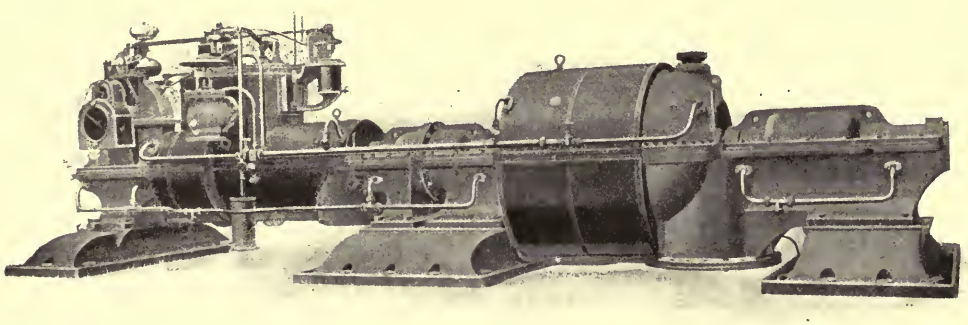


FIG. 192.—Parsons' Steam Turbine (standard large sizes).

Having now particularly described and ascertained the nature of my said invention, and in what manner it is to be performed, I declare that what I claim is :—

In steam turbines, the combination of the turbine wheel with a steam supply, the cross sections of which increase regularly near to the turbine wheel and in the direction of the same, substantially as and for the purpose specified."

That's all. And note the simple explicit drawings. It is safe to say that if this invention had originated in New York, the specification would have filled some pages of this book and the claims filled more space than De Laval's whole specification, while the possibilities offered in the way of drawings are unlimited. A whole series of possible and impossible turbines would have been shown, with furnaces, chimneys, boilers, gas works, superheaters, condensers, and every conceivable detail carefully drawn.

De Laval accepted the high speed necessary with a single wheel, and formed his nozzle on scientific lines, so that the full velocity may be attained as already described and illustrated under "Steam Jets."

Parsons had already shown that elasticity in the bearings of the turbine wheel shaft was necessary to damp vibrations, and allow the wheels to approach a revolution round the centre of gravity; but De Laval aimed at higher peripheral speeds, and so, instead of damping vibration, he arranged his wheel on a flexible shaft so that the shaft could bend and allow the wheel to find its own centre of revolution. This flexible

shaft was at first made with spring actions, but, owing to the fact that the torque of a wheel at these high revolutions is small, the shaft can be made thin enough of itself to bend sufficient for the purposes. The first specification is as follows :—

The invention consists of an improved arrangement for reducing to a minimum the reactions on the bearings by rapidly rotating bodies, arising from their imperfect balancing. This is accomplished by fixing the rotating body either directly or indirectly on a tube or sleeve which is fixed on and around a shaft, which tube is made elastic or yielding in some suitable manner, so that the rotating body is rendered self-adjusting or self-balancing.

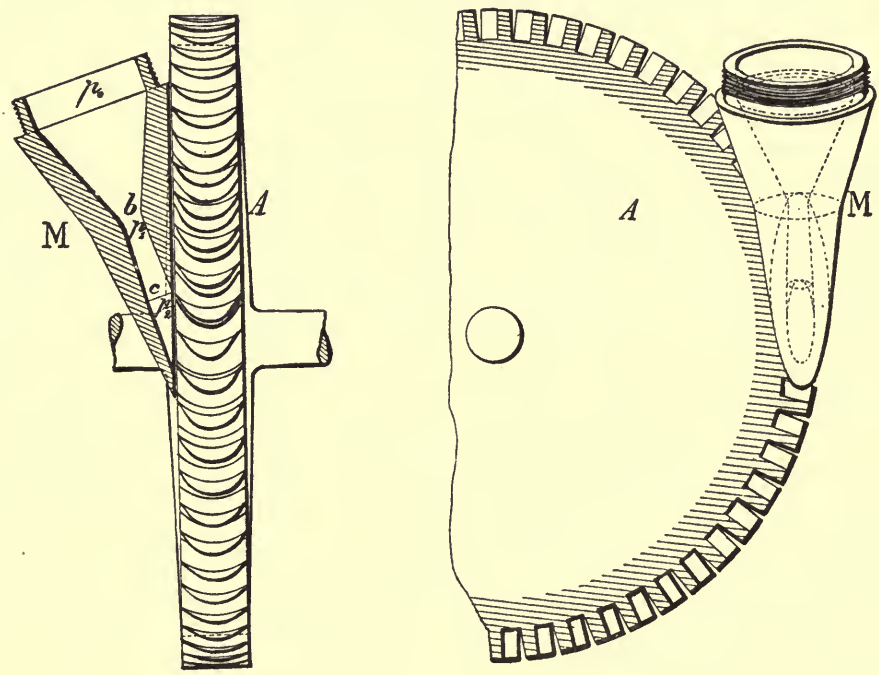
Two constructions of this arrangement are shown—

Fig. 194 being a part sectional elevation of one construction, and

Fig. 195 a like view of another construction, showing a transverse sectional view on the line *cd*, and a transverse sectional view on the line *ab*.

Similar letters of reference indicate corresponding parts.

In Fig. 194 the body A is fixed on the boss B, which again is made fast upon the



Front View. Side Elevation.

Fig. 193.—De Laval Wheel and Nozzle.

central part of the tube or sleeve C, the latter being fastened on both ends D and E to the shaft F. This shaft is shown supported at each end in bearings G and H in the usual way, and the motion is imparted to the system by means of the driving pulley K. A portion of the shaft F between D and E has been turned down, thus leaving an annular space between the tube C and the shaft F. A helical slit *b* is cut in the tube C between its central portion and each end, so that the central portion occupied by the body A may, while rotating, assume such position relative to shaft F as may be required to secure an even rotation, which saves wear and tear in the bearings.

In Fig. 195, where the body A is directly fixed upon the thicker central portion of the tube or sleeve C, this tube is made elastic or yielding as required, either by turning down the tube until its thickness is considerably reduced (see transverse section through *ab*), or the tube, being somewhat thicker (see transverse section through *cd*), is provided with longitudinal slits *a* by means of which the required elasticity is obtained.

The bearings for the ends G and H of the shaft have in the present cases been shown fixed. In some instances, however, it may be desirable that these also should be able to yield to a slight extent. In such cases Seller's bearings should be used, pivoted bearings, or bearings surrounded with indiarubber rings like the top bearing of the cream separators. Other constructions of self-adjusting bearings may also be used to give the requisite adjustment.

In a subsequent patent De Laval shows how to apply this flexible shaft with special jointed bearings to other rapidly moving heavy rotating bodies, so that a balance is obtained. This and the previous patent display true engineering science, in which a desirable result is achieved by natural means. The old-time engineers, when they first introduced moderately high speed engines, tried to balance the cranks by counterweights and other means, bringing the inertia of the revolving parts into opposition; but they relied upon some tons of cast iron, stone, and concrete, and ten or twelve huge foundation bolts, after all, to prevent vibration. They did not cure vibration; they only made it less in amplitude by mere force, and the internal strains were greatly increased. The early engines were kept from violent vibration by brute force, not by any scientific means. Nowadays, engineers do not depend on foundations for smooth working: a good engine well designed and balanced should run full speed without vibra-

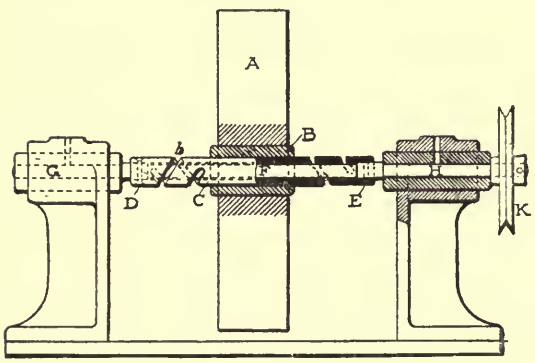


FIG. 194.—De Laval Flexible Shaft.

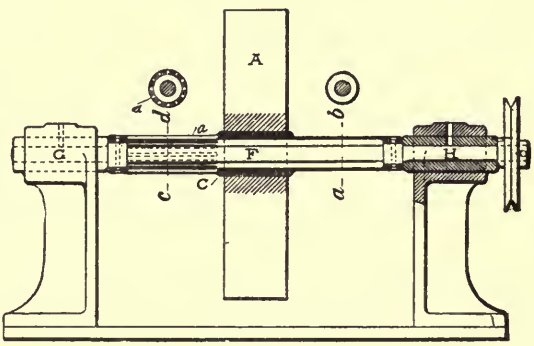


FIG. 195.—De Laval Flexible Shaft.

tion when standing upon a foundation without any holding down bolts. A high speed engine which requires to be held down to a huge foundation by heavy bolts is an engine radically wrong in design.

A theory of the flexible shaft has been given by Mr. Sandford A. Moss in *Power*, which may be summed up as follows:—

The popular explanation of the phenomenon is that the flexible shaft allows the disc to "rotate about its centre of gravity." This is not true, however, since in all cases a disc slightly out of balance rotates so that the shaft is slightly deflected and the centre of gravity is away from the centre of rotation. The outward centrifugal force due to this eccentricity of the centre of gravity is just in equilibrium with the inward pull due to the deflection of the shaft. Of course this deflection causes a rotating reaction at the bearings, which tends to vibrate them, and which usually gives trouble in high speed machinery. This is because the greater the speed the greater the centrifugal force, and ordinarily the greater the shaft deflection required to balance it. The increase of deflection throws the centre of gravity out farther, which still further increases the centrifugal force. Hence ordinarily the reaction on the bearings due to the shaft deflection, which is of course equal to the centrifugal force, increases rapidly with increase of speed. However, it is a remarkable fact that, after a certain critical speed has been reached, the shaft deflection and the reaction on the bearings actually decrease as the speed increases, in order that equilibrium may be maintained with the centrifugal

force due to the corresponding eccentricity of the centre of gravity. The eccentricity is less than the deflection, instead of being greater as ordinarily. The more flexible the shaft the less this "critical period."

By using a very flexible shaft and running it at a much greater speed than the critical speed, the deflection required to maintain equilibrium with the centrifugal force due to the corresponding eccentricity of the centre of gravity becomes very small indeed, and the corresponding rotating reaction pull on the bearing is almost imperceptible.

The critical speed in radians per second is found to be $\psi=\sqrt{\frac{Kg}{M}}$, wherein ψ is the natural period of vibration of the shaft if deflected and let go or struck in the middle, M is the weight of the disc in lbs., and K the force required to deflect it unit distance.

In revolutions per minute this becomes $\frac{60}{2\pi}\sqrt{\frac{32.16\times 12k}{M}}$, or $187.56\sqrt{\frac{k}{M}}$, where k is load in lbs. required to deflect the shaft 1 inch and M is the weight of the disc in lbs.

Now M/k is the actual deflection of the shaft in inches, produced by the weights of the disc; hence the critical speed is $\frac{187.56}{\sqrt{d}}$ revolutions per minute, where d is the deflection of the shaft in inches produced by the weight of the disc.

For a steel shaft of the same diameter throughout, with two bearings either swivelled or so loose that the shaft may be considered as merely supported by them, and for a single mass fixed at any point between the bearings, the formula reduces to the following: Let l be the distance between bearings in inches, M the weight in lbs. of the rotating mass, a its distance from either of the bearings in inches, and d the shaft diameter in inches; then the critical speed at which the maximum vibration occurs, in revolutions per minute, is $387{,}700\frac{d^2}{a(l-a)}\sqrt{\frac{l}{M}}$. This also gives the period of a natural vibration. If there are two rotary masses, there are two distinct vibration periods and critical speeds. Now as to the efficiency of such wheels, as mentioned previously, it is important that as much as possible of the kinetic energy of the steam jet issuing from the nozzle should be taken up by the turbine wheel, and thus transformed into mechanical energy. The angle between the nozzle and the plane of rotation of the wheel is 20 degrees, and in order to obtain the maximum efficiency the peripheral speed of the turbine wheel, *i.e.* the linear velocity of the buckets, should be 34 per cent. of the velocity of the steam. The absolute velocity of the steam leaving the buckets is then 34 per cent. of the initial velocity, and the energy absorbed by the turbine wheel is 88 per cent. of the kinetic energy of the steam.

If, for instance, the speed of the steam entering the buckets of the turbine wheel is 4000 feet per second, the speed of the steam leaving the buckets should be 1360 feet per second, and the number of horse-power per lb. of steam $\frac{4000^2-1360^2}{2g\times 550\times 3600}=11$; and the steam consumption per theoretical horse-power $\frac{2g\times 550\times 3600}{4000^2-1360^2}=9.1$ lbs.

The steam nozzles are placed in very close proximity to the buckets of the turbine wheel, in fact the distance is only 2 millimetres, or about $\frac{1}{16}$ of an inch, and consequently there is practically no loss of velocity between the steam jet leaving the nozzle and entering the buckets of the turbine wheel, nor leakage of steam.

The speed of the turbine wheel, which for a velocity of the steam jet of 4000 feet per second ought to be about 1880 feet per second, or about 21 miles per minute, is, however, much lower, for several practical reasons. At the present time the peripheral speed of the De Laval turbine wheel does not exceed 1380 feet per second, which should

make a steam consumption of 9.8 lbs. per theoretical horse-power. The following table gives the speed of some types of turbine wheels :—

TABLE XVI.—SPEEDS OF THE TURBINE WHEELS.

Size of Engine.	Middle Diameter of Wheel.		Revolutions per Minute.	Peripheral Speed. Feet per Second.
Horse-power.	Millimetres.	Inches.		
5	100	4	30,000	515
15	150	6	24,000	617
30	225	$8\frac{7}{8}$	20,000	774
50	300	$11\frac{3}{4}$	16,400	846
100	500	$19\frac{3}{4}$	13,000	1115
300	760	30	10,600	1378

Fig. 196 shows the stresses in a wheel for a 50 horse-power steam turbine.

As may be seen by this diagram, the wheel is so constructed that both P the radial stress and S the tangential stress have their largest value at the circumference of the wheel, just where the buckets are fixed. Consequently the wheel is not made of uniform strength, but is strongest at the heavy part, that is, in the centre.

It will be seen from Fig. 196 that the tangential stresses S in the boss of the wheel increase as they approach the hole in the centre of the wheel. These stresses would be still greater on the larger sizes of wheels, and in order to avoid these greater stresses the larger wheels are made without any hole through the centre, but the shaft is made in two pieces fixed to the wheel by flanges and screws. Fig. 197 shows the arrangement of small and medium turbine wheels; and the arrangement of the larger turbine wheels is shown in Fig. 198.

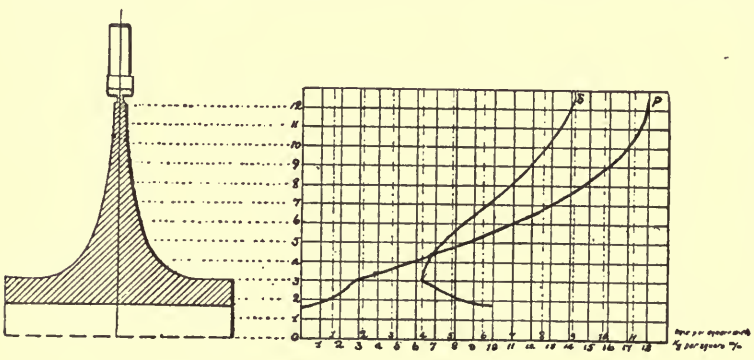


FIG. 196.—Curves of Stresses.

The part of the turbine which makes it possible to run the turbine wheel at its enormous speed is the "flexible shaft." The shaft on which the turbine wheel is mounted has bearings on each side of the wheel at a good distance from it, and the shaft is consequently flexible and can allow the wheel to swing a little in its plane of rotation. No matter with what nicety the turbine wheel may be turned and balanced, it is practically impossible to bring the centre of gravity of the wheel exactly into the geometrical centre round which the wheel revolves.

The flexible shaft and the turbine wheel are so proportioned that the settling of the wheel takes place very quickly, and the critical speed is from $\frac{1}{5}$ to $\frac{1}{8}$ of the standard number of revolutions of the wheel. Of course the turbine wheels are very finely balanced, and the settling of the wheel is therefore scarcely perceptible. It is the flexible shaft that serves to transmit the power of the turbine.

The diameter of the shaft is, on account of the high speed, very small, and it is therefore easy to make it flexible. The shaft of the 300 horse-power turbine wheel has a diameter of 34 millimetres, or $1\frac{5}{16}$ inch, and that of the 150 horse-power wheel 25 millimetres, or 1 inch : no larger diameter is required.

The normal speed of the turbine wheel is too high for direct driving of ordinary machinery, and it is therefore reduced by means of gearing. This gearing is made on the double helical system, and machined with the greatest care and accuracy, as is necessary on account of the high speed. The speed of the gearing, that is the linear velocity of the teeth, is about 1000 feet per second. The pinion is made of hard steel (in one piece with the shaft), and the teeth of the gearing wheels are cut in a somewhat softer steel than the pinion.

All the revolving parts of the turbine are most carefully balanced, and the parts mounted on the shafts are centred by tapers. The bearings of the slow speed shafts are lubricated by rings, as is usual in this class of machinery. The journals of the flexible shaft are oiled by sight feed lubricators. The bearings, which are all made as interchangeable bushes, are lined with white metal, and there is practically no wear on them if the machine is well lined up and properly mounted from the beginning.

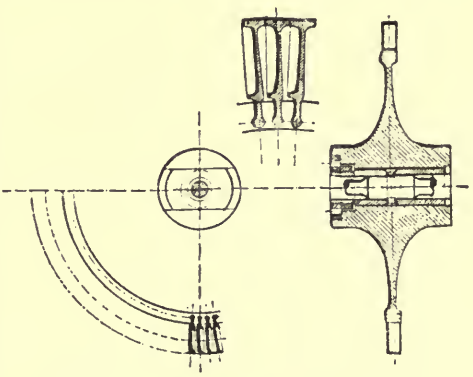


FIG. 197.—Arrangement of Small and Medium Turbine Wheels.

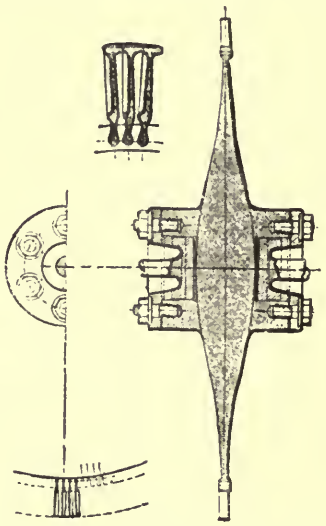


FIG. 198.—Arrangement of Large Turbine Wheels.

The turbines are generally fitted with more than one steam nozzle, and these are arranged at intervals in a ring in close proximity to the turbine wheel, receiving the steam from a steam chest in the turbine case. Each nozzle is usually provided with a

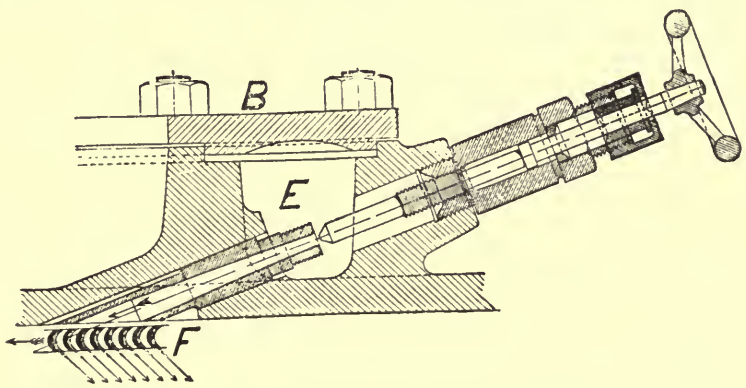


FIG. 199.—Valve and Nozzle of De Laval Turbine.

shutting-off valve, so that any nozzle can be closed or opened at any time. This arrangement is of considerable advantage, as when the turbine is working at reduced load some of the nozzles may be closed and a high efficiency of the machine maintained, even although it is not working at full load. This will be more plainly understood from the tests of steam consumption, which will be noticed subsequently. The arrangement of a nozzle and its valve will be seen from Figs. 199 and 200.

As may be seen from the foregoing table, the peripheral speed increases with the size of the wheel, and the larger the diameter the higher also is the peripheral speed. The 300 horse-power turbine wheel runs with a peripheral speed of 1378 feet per second in the middle of the buckets ; the outside diameter of this wheel is 800 millimetres, or 31½ inches, and the circumferential velocity of the wheel is 1450 feet per second, or more than 16 miles per minute. At this speed the wheel would travel round the equator of the earth in 25 hours.

On account of the peripheral speed of the turbine wheel not being so high as it theoretically ought to be, there is, particularly at high admission pressure and good vacuum, a slight impact when the steam enters the buckets. This is, however, allowed, for practical reasons, and the energy due to the loss of speed by this impact is not entirely lost by the turbine, as will be seen later.

One advantage of the action principle of the turbine is that the turbine wheel can revolve quite freely in the casing. This is an essential feature of the machine, and, moreover, it would not be possible to run at the speed required should a tightening be necessary round the turbine wheel. The wheel does not touch anywhere, and all the steam on emerging from the nozzles must pass the buckets of the wheel, as the radial length of the buckets is always larger than the diameter of the steam jet. There is consequently no possibility of any steam leaking through the turbine, but it must of necessity pass the buckets and deliver its energy to the turbine wheel.

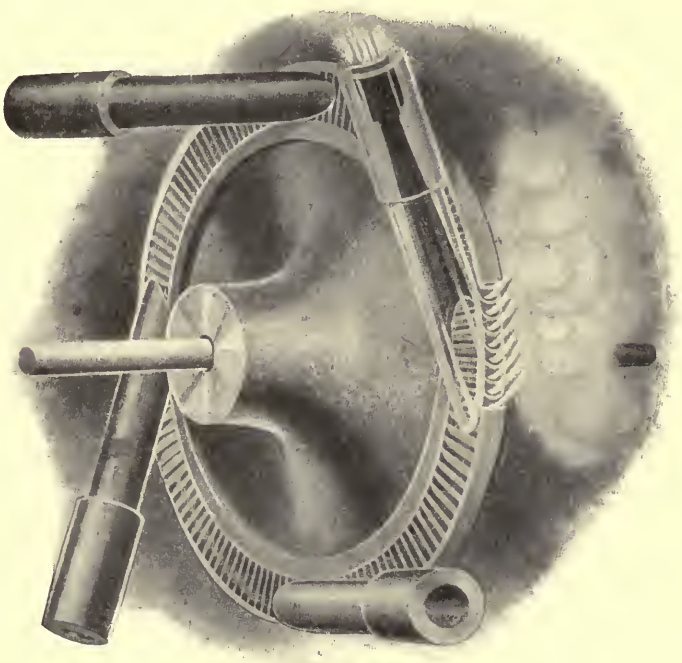


FIG. 200.—De Laval Wheel and Nozzles.

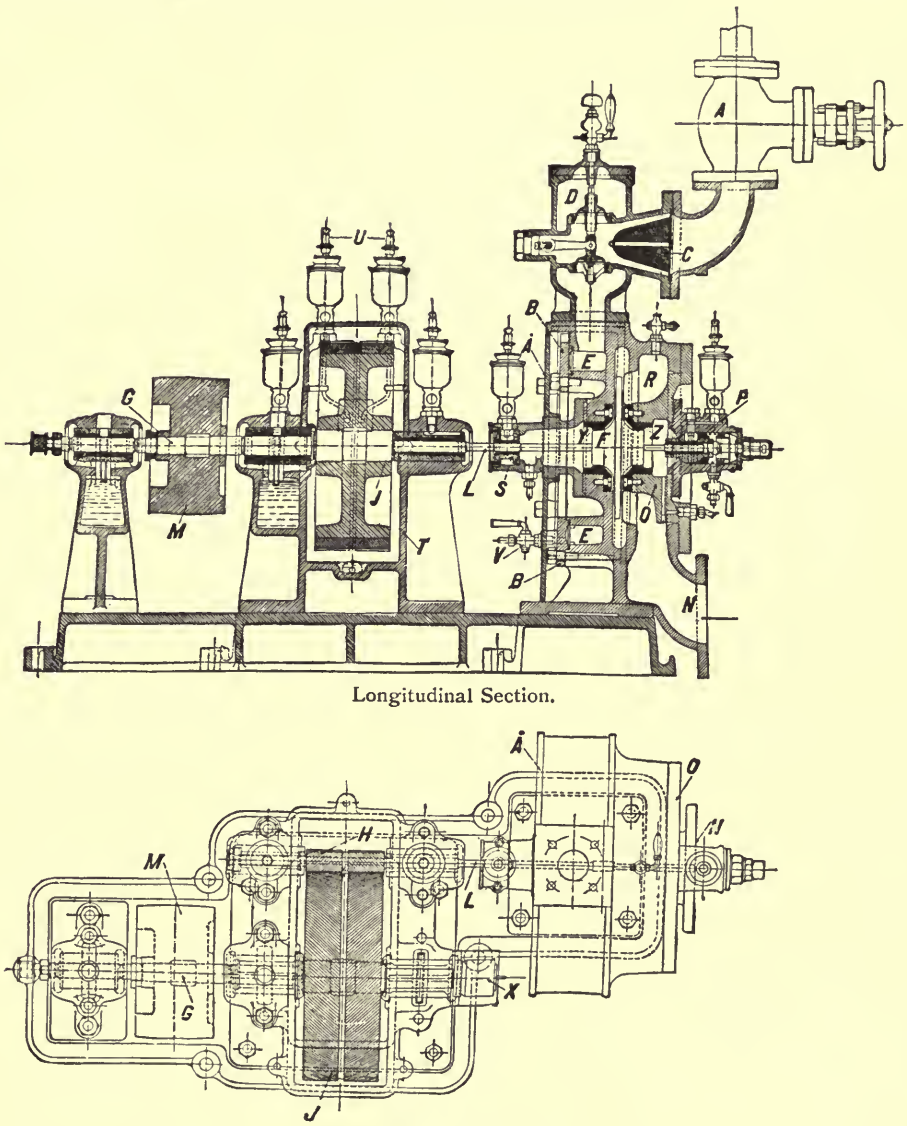
The high peripheral speed which, as previously seen, is necessary in order to obtain a good efficiency has been obtained by allowing the turbine wheel to run at a very high velocity. A reference to Table XVI. will also show that the number of revolutions is much higher than the speeds formerly used in practical engineering.

The wheel is made as a solid disc, on the circumference of which the buckets are dovetailed in, each bucket being made and fixed separately to the wheel. The buckets consequently load the circumference of the wheel with a radial force when the wheel is revolving. The amount of this force may be understood when it is mentioned that the centrifugal force on the bucket of a 300 horse-power turbine wheel, which bucket weighs 250 grains, is 15 cwts. when the wheel is running at its standard speed.

The stresses in the wheel are tangential and radial ; and if we call the radial stress P and the tangential stresses S, it is evident that both P and S vary with the radius R.

Further, these stresses depend on the axial thickness of the wheel in each place, and they also affect one another.

Before the admission steam can enter the steam chest and pass from thence to the nozzles it is regulated by the governor valve, which in its turn is controlled by the centrifugal governor of the machine. The governor valve is a balanced double-seated valve, connected with a link motion to the centrifugal governor.



Longitudinal Section.

Sectional Plan.

FIG. 201.—De Laval Turbine.

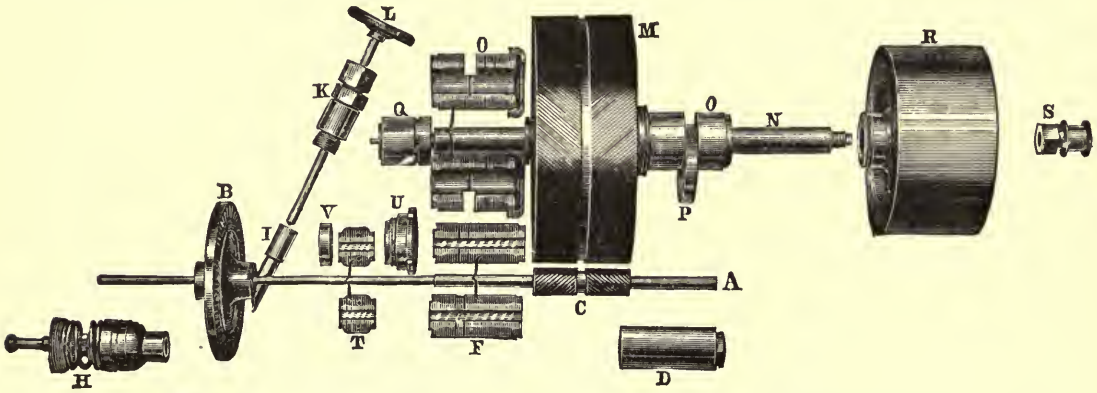
The general arrangement of the machine will be understood from Fig. 201, which shows a turbine in section and plan, and Fig. 202, a dissected turbine.

The speed of the turbine is regulated by a very sensitive centrifugal governor, mounted horizontally on the end of the gear-wheel shaft. The moving parts of the governor work practically without friction, and it is therefore very quick and powerful. This governor is very simple, although the construction may seem peculiar, and its

dimensions are very small on account of the comparatively high speed at which it works. Fig. 203 gives an idea of the construction of the governor.

The variation of speed between full load and no load is nearly 1 per cent.; the variation is from 2 to 3 per cent. generally.

The standard sizes of steam turbines can work with any steam pressure between 50 and 200 lbs. per square inch, and either with or without vacuum. The only parts of the machine which have to be arranged to suit the admission pressure and the pressure in



A Turbine shaft
B Turbine wheel
C Pinion
D End bush
E Safety bearing in turbine box
F Middle bush in two parts
G Safety bearing in box covers
H Ball bush with adjusting spring
I Steam nozzle
K Stuffing box with stop valve to nozzle
L Wheel to valve spindle
M Gear wheel
N Gear wheel shaft
O Gear wheel shaft bushes in two parts
P Lubricating ring
Q Centrifugal governor
R Driving pulley
S Stop nut with pulley for connecting to tachometer
T Tightening bush in two parts
U Adjusting nut with spring
V Friction gland

FIG. 202.—Loose Parts of Steam Turbine.

the exhaust are the steam nozzles, which have to be shaped according to the amount of expansion of the steam. The nozzles are made interchangeable—all other parts do not alter with the pressure,—and the machine can consequently work with any pressure between the above limits, if only the turbine case is provided with suitable nozzles. The turbine can also be arranged with nozzles for running both condensing and non-condensing: this is very handy and convenient, particularly if the turbine drives its own condensing machinery, direct or electrically.

As the question of economy became of more importance, the size of the wheels and also the number of revolutions in the larger unit of machine are so proportioned that, with increasing unit, the velocity of the vanes of the wheel approached more closely to what it theoretically ought to be. The stresses are largest in the circumference of the wheel, and the sections of the wheel were so proportioned that the stresses increased with the radius. This was done in order that the wheel might be weakest in the circumference where the vanes were fixed, and in order to be still more certain on this point a recess was turned in this outer portion of the wheel. Should a wheel burst on account of too high a speed it gave out at the recess, the vanes became detached, and the steam could not longer drive

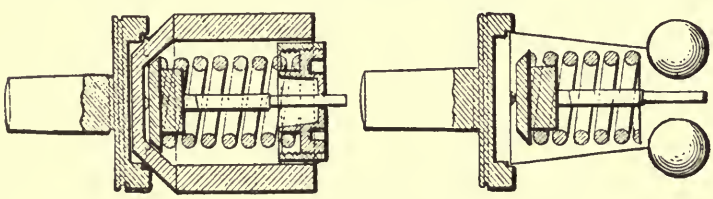


FIG. 203.—Governor of De Laval Turbine.

the wheel. The buckets of the detached parts of the wheel were so light that they could do no damage, and the machine only stopped running. With this type of wheel the heavy central part had never burst. Indeed it would be a very serious matter if the heavy part ever became detached from the shaft. When the calculations for a new wheel were completed, the material of which the wheel had to be made was tested in order to see whether its strength corresponded to that on which the calculations were based. If agreement existed, the wheel was then made, and run until it burst at the periphery. Experience had proved that the speed at which the breaking of the wheel took place could be calculated beforehand, and, if the speed obtained by experiment accorded with the theoretical result, the wheel was adopted as a standard for that particular type for practical use. The wheels were generally proportioned so that breaking would take place when the wheel was running at about double the number of revolutions required in actual work; consequently it could hardly happen in practice that the machine increased so much in speed that there would be any danger from the wheel breaking at its circumference. In order to keep the stresses down in the central parts, the wheels were made very thick towards the axis. If a wheel of the same thickness throughout were used, the stresses would be very great at the centre, and this was a matter to avoid in high speed machinery. It was best to do away as much as possible with the drilling

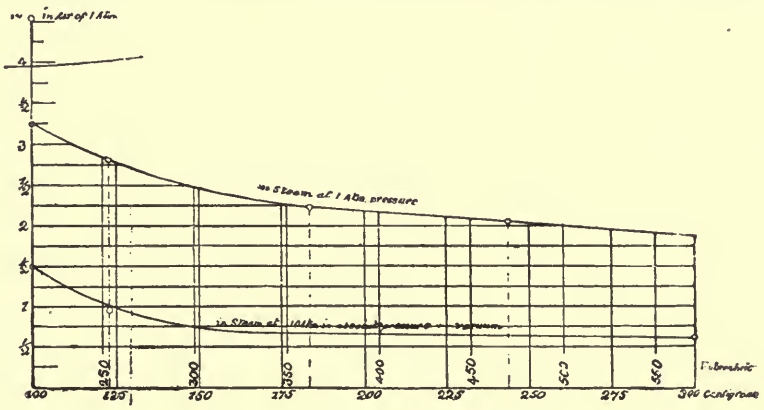


FIG. 204.—Skin Friction Curve.

of holes for bolts, etc. in the boss of a wheel, as holes considerably weakened the central parts, and it was very often the boss which had to keep the wheel together. As to the resistance to which the revolving wheel was subjected from the surrounding medium, this depends partly on the skin friction, and partly also to eddy making. It was found in practice that the resistance was almost exactly proportional to the density of the surrounding medium, and that it increased approximately with the fifth power of the diameter and the third power of the number of revolutions. It was therefore evident that the thinner the medium which surrounded the wheel, the less would be the resistance offered to its motion, and this would be plainly understood from the curves in Fig. 204. The resistance was less in saturated steam than in air of the same pressure, and it decreased with increasing vacuum. A 150 horse-power turbine wheel was subjected to a resistance of 35 horse-power when running in steam of 1 atmosphere absolute pressure, but if it were run in a vacuum of 28 inches of mercury—2 inches of mercury absolute pressure—the resistance would be decreased in about the same proportion, and would be $\frac{2}{30} \times 35 = 2\frac{1}{3}$ horse-power, a gain of $32\frac{2}{3}$ horse-power. The velocity of the steam jet into the vacuum was, moreover, higher than the velocity of outflow into the atmosphere, and both these circumstances made it essential that the turbine machinery should be run under vacuum. From Fig. 204 it was also evident that the resistance was less in superheated than in saturated steam, and that it decreased with the amount of superheating.

The frontispiece of this volume illustrates two fans driven by two of these turbines installed at Romford Gas Works.

Plate VI. illustrates the smaller sizes set up with reducing belt gearing all self-contained for driving small powers : this illustrates a 5 horse-power set.

And Plate VII. illustrates the large Parsons' turbine with alternator attached. The frontispiece and Plate VI. are from Messrs. Greenwood & Batley, who are the British representatives and manufacturers of De Laval turbines.

Before passing on to notice the few turbines introduced since the great success of Parsons and De Laval, it may be of interest to notice another effect of high velocities in rotating bodies, namely the gyrostatic effects : the inertia of a body opposing any force which tends to set it in motion, or to alter its motion while moving, either in direction or velocity.

The gyroscope illustrates these properties beautifully. The gyroscope usually consists of a fly-wheel mounted on centres inside of a ring, so that it can be spun round at a high velocity. It is sold as an interesting toy, but some of the most intricate mechanical problems can be found in its theory. It is also of use for showing the earth's rotation.

A bicycle wheel may be used to demonstrate a few effects of the gyroscope. The rear wheel is usually easily removed with its spindle. If this is done and two people hold it up, one grasping each end of the spindle, and the wheel be rapidly spun round, it will be found to strongly resist any attempt to twist the shaft round, out of the line it was held in. One of the persons may let go his end of the spindle, the wheel will not fall, and the other person will not have to exert any more support than that required to keep the weight of one end up. In fact, the wheel, while rotating in a vertical plane, can be suspended by a string tied to one end of the spindle. The spindle will not drop out of the horizontal position, even although supported at one end only.

If a gyroscope be taken between the two hands, or the spinning bicycle wheel between two persons, and the wheel turned right over, so that the spindle is turned end for end, a considerable resistance will be offered by the wheel to this operation. The mathematical explanation for this resistance is complicated, but the simplest way to look at it practically is to consider the direction of the motion of the wheel, first before it is turned over, and after. Suppose a person looking on one side of the wheel when it first spins round, that he sees the direction of rotation is from left to right, that is clockwise, he will find, on turning the spindle end for end against the resistance of the wheel, that the motion or direction of rotation has from his point of view reversed. The resistance offered by the wheel is the resistance which a body in motion offers to reversal of the direction of the motion, just as a jet of steam or water resists reversal in the bucket of a turbine.

In fast running machinery like steam turbines, any alteration of the plane of revolution is accompanied by resistance which bears on the shaft and bearings. In ships the shaft is horizontal, so that any quick turning of the ship will throw pressure on the turbine bearings resisting the turning round of the ship. Rolling of the ship would not have any effect, for the plane of rotation would not be altered by a ship's rolling with the turbine shaft fore and aft. Pitching would have but a slight effect, for the angle of movement is small. A rapid deviation of the ship at full speed from her course would have most effect ; and in ships of high speed and light weight, such as torpedo-boats and destroyers, with powerful engines and powerful steering gear, it is quite possible that a sudden swerve would throw a dangerous amount of strain on the turbine bearings, and might bring about a rupture of the structure.

A rolling ship with the turbine shaft athwart the ship would also be subjected to straining, for the rolling would considerably and quickly alter the plane of rotation.

Experiments with gyroscopes are now mostly made with electrically propelled wheels, the motion being steady, and prolonged as long as we please.

There are three or four turbines which have been introduced into practice, while there are some three or four thousand which have not been reduced to practice. Since Parsons and De Laval established the practicability of steam turbines fitted with proper bearings, guide blades, and wheel shafts and other things, all designed with strict regard to the elastic fluid and high velocities necessary, there has been the usual avalanche of patents poured into the patent offices. About fifty patents were granted in 1901, and now in 1903 only four or five of them have ever since been heard of.

I have tried to go through all the patents since 1890 up to the latest published. Ninety-nine out of every hundred are utterly sterile of any original ideas. Many are not so far advanced as Pilbrow and Wilson were fifty years ago. The fact is, when we add Parsons' improvements and De Laval's improvements to those of Pilbrow and Wilson, we have the whole state of the art and science of steam turbine construction at the present moment clearly founded on the published specifications of these four inventors.

The master patents having expired by lapse of time, the vast number of new patents can only cover some detail, such as bearings, valves, packing, special nozzles or blades, so that all recent turbines are either Parsons' or De Laval's, with some subordinate feature altered or changed for another.

Some of them, if equally skilfully designed and equally well made, would no doubt give as good results, but that "if" is of a large order.

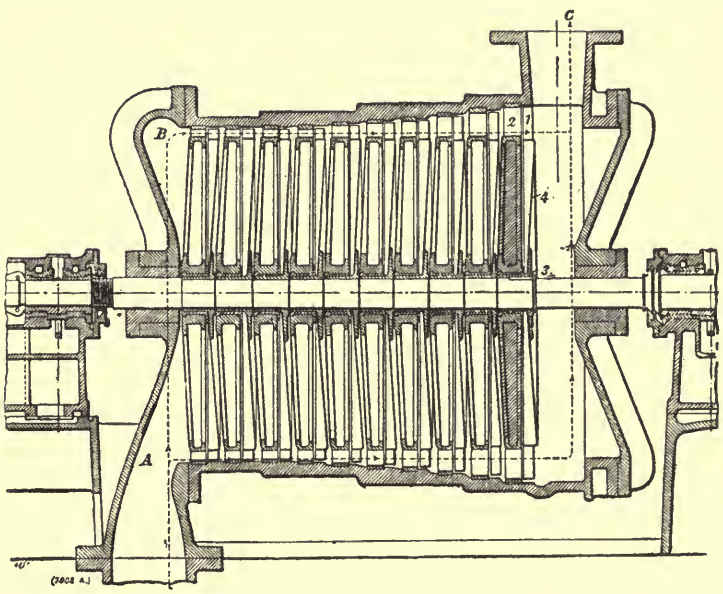


FIG. 205.—Rateau Turbine.

The Rateau turbine is essentially a number of De Laval turbines in series with one common shaft, and is one of the first rank, designed by Professor Rateau. Its scientific points are above criticism ; and, made with that mechanical skill which French mechanics display in all their work, it has rapidly assumed an important position. In *Engineering* a brief description of it as given by Professor Rateau himself is as follows :—

The multicellular turbine in question is constructed by Messrs. Sautter-Harlé & Co., engineers, of Paris, and consists, as shown in Fig. 205, of a series of rotating wheels keyed on the turbine shaft, and separated by circumferential diaphragms held in grooves inside the turbine casing. The latter is in two parts, the section being on a plane which passes through the axis of the turbine. This allows the accurate fitting of the lower half of the system, which is then completed by putting in place the top half of the casing.

The wheels consist of buckled steel plates, conical in shape (Fig. 205), on the periphery of which are riveted nickel-steel blades, which are further held in place by a band. The fixed blades of the distributors (guide blades) are fitted on the periphery of the diaphragms (see 2, Fig. 205), opposite the wheels. On Fig. 205 the last diaphragm is hatched in order to throw it up more clearly.

The shaft runs through the diaphragms in bushes of anti-friction metal, with but little play—originally two-tenths of 1 millimetre. The shaft, however, makes its play

when this is insufficient. There is no inconvenience attached to the friction of the shaft in the anti-friction bushes. Leakages of steam outside the distributors can only take place through this small windage, the total section of which is comparatively a very small one, owing to the small diameter of the shaft. The space between the wheels and the fixed parts is from 3 to 6 millimetres .118 inch to .236 inch; there is thus no fear of their coming into contact. Should this happen, however, and a number of blades get broken, they could easily be repaired, even on board ship, as each blade can be removed individually and be replaced by another, fixed by one rivet only.

The steam flows through the diaphragms and wheels from the inlet at AB to the exhaust G, its pressure decreasing progressively, each successive wheel working under a low fall in pressure. The number of wheels varies from three to four only to thirty and over.

The foregoing shows that all loss in efficiency through leakage between the fixed and movable parts has been prevented in a marked degree. It has been stated that with this system of turbine the friction of the wheels in steam leads to a notable loss. This is

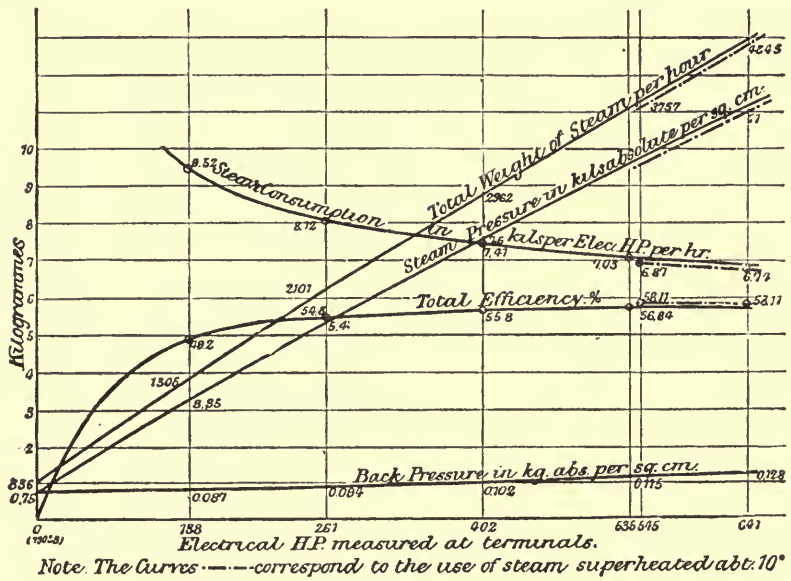

Kilogrammes
Steam Consumption
Total Weight of Steam per hour
Steam Pressure in kils absolute per sq. cm.
kals per Elec. H.P. per hr.
Total Efficiency %
Back Pressure in kg. abs. per sq. cm.
Electrical H.P. measured at terminals.
Note. The Curves ·—·—· correspond to the use of steam superheated abt. 10°

FIG. 206.—Results of Tests on Rateau Turbine.

not the case; and in actual practice, with a turbine the dimensions of which have been carefully calculated, the loss by steam friction is only 3 to 4 per cent. of the power of the machine.

The steam consumption is also very moderate, being, it is believed, lower than that of any other similar type of engine. These turbines are calculated with great precision, by formulæ based on theory and completed by experiments on the flow of steam; the difference between the calculated figures and the results obtained at the trials does not generally exceed 2 per cent., thus confirming the formulæ on which the *études* were first made.

The efficiency of the turbines is shown in the diagram, Fig. 206, which gives the results obtained at trial runs made with the first of three turbines, coupled to a continuous current dynamo, supplied by Messrs. Sautter-Harlé & Co. for the generating station of a mine at Penarroya, in Spain. The only motive engines in this station are the three steam turbines in question.

From the diagram, Fig. 206, it will be seen that the consumption of the turbine and dynamo under full load or 500 horse-power—the inlet pressure being 8.5 kilogrammes

per square centimetre or 120 lbs. per square inch, and the back pressure at the condenser .115 kilogramme per square centimetre (1.63 lb. absolute)—was 7.03 kilogrammes or 15.5 lbs. of steam per electric horse-power hour ; and under an overload, at 640 horse-power, with 11 kilogrammes or 156.5 lbs. per square inch and .128 kilogramme or 1.82 lb. absolute counter pressure at the condenser, the steam consumption was only 6.74 kilogrammes or 15 lbs. of steam per electric horse-power hour. These are good results for a 500 horse-power engine, seeing especially that the vacuum at the condenser was not very high. With a high vacuum, such as the Hon. Charles A. Parsons can so readily obtain, say 0.06 kilogramme or 0.86 lb. absolute back pressure at the condenser, the above figures of steam consumption would be reduced by 10 per cent. at least, and would be 14 lbs. per electric horse-power hour normally, and 13.4 lbs. with an overload.

The diagram, Fig. 206, shows further that the total efficiency of the set, which rises to 58 per cent. at full load, remains approximately constant down to below half load ; further, the total consumption of steam—the dynamo remaining excited and being self-exciting—does not exceed 336 kilogrammes (740 lbs.) per hour running empty, being thus 10 per cent. lower than at full load. M. Rateau believes that no other engine gives such favourable results.

It is an impulse turbine, but the distinction between impulse turbines when put in series order and so-called reaction turbines, as Mr. Parsons has very properly pointed out, is the difference between tweedledee and tweedledum, when the fluid is steam. M. Rateau denies this, and refers to hydraulics to show the difference. But water and steam are two very different fluids. While they act similarly in some respects, they differ very materially in others, due to the fact that one is elastic and the other is not.

OTHER STEAM TURBINES

There is the Stumpf turbine, a modification of the De Laval, with a wheel more like a Pelton wheel, and nozzles all around.

The Seger turbine is like Pilbrow's 2-wheel machine shown in Fig. 161, and my own wheels shown in Fig. 163. A feature of this machine is the reduction of the speed by belt gearing.

There is also the Schulz and Curtis steam turbines ; but a perusal of the specifications reveals nothing of much novelty in either, but we shall have occasion to examine them.

The Westinghouse Company's steam turbine is a modification of Parsons'. At an early date this company purchased the American rights and licence for the British. The Thomson-Houston Company in this country supply the Curtis turbine as modified by the engineers of the General Electric Company of America at Schenectady.

German turbines are known as those of Professor Stumpf, and Schulz. French turbines as designed by Rateau are made by Sautter-Harlé & Co.; while De Laval turbines are made in many countries.

These turbines we notice here more for the purpose of showing the small details which distinguish them from the earlier and more original turbines.

The construction of high speed wheels with curved blades has always been somewhat a difficulty. We have seen how Parsons and De Laval make their wheels. E. Seger, of Stockholm, in his patent of 1894, No. 4611, shows a simple method for forming wheels for high velocities ; he employs two wheels side by side, like Pilbrow's, Fig. 161, running in opposite directions.

The following are the instructions for manufacturing the wheels, in which the vanes or buckets are called paddles :—

Referring to the illustrations—

Fig. 207 is a vertical section of two turbine wheels arranged one above the other.

Fig. 207A is a front elevation of the same with the wheel ring partly removed.

Fig. 208 illustrates the form of the paddle blanks.

Fig. 208A is a sectional view of part of a turbine wheel with an inserted paddle, but without the outer wheel ring.

Fig. 209 is a side elevation of a part of such a turbine wheel with paddles, illustrating two different steps of the operation of fastening the latter.

Fig. 209A shows a part of such a turbine wheel with the paddles fixed by riveting, and

Fig. 209B shows the outside of a part of the wheel body with notches for the paddles, some of the paddles being inserted.

The paddle blanks *a*, which may be punched out of thin sheet-iron plates or of

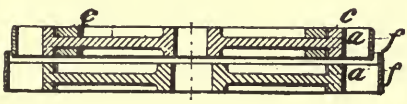


Fig. 207.

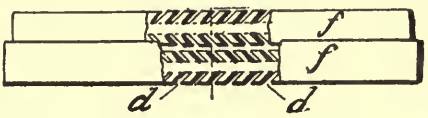


Fig. 207A.

plates of some similar material, have preferably a substantially rectangular form (Fig. 208), with a notch *b* in its undermost side. The lower corners of this rectangle may be cut off (as is illustrated by Fig. 208), or they may be formed in some other way in such a manner that they do not overlap the edge of the wheel body when the undermost part of the paddle blank is bent, as will be hereinafter further described. The body of the wheel has an I-shaped section (Fig. 207), and at the two edges of the wheel ring *c* there is a number of notches *d* (Fig. 207A), of which those at the one edge converge towards those of the other edge, without, however, meeting the same. Into the said notches the flaps *x* on both sides of the notch *b* are inserted, the paddle blank being previously somewhat curved. If the turbine is a compound turbine, consisting of two turbine wheels, the one above the other (as illustrated in Figs. 207 and 207A), the notches *d* of the upper wheel must have a sharper convergence than those of the lower wheel. After the insertion of the paddle blanks in the notches the ends of the flaps *x* projecting beyond the inside of the wheel ring are bent against the inside of that ring (Fig. 209), thus causing the paddles to be securely held in position even at the greatest velocity of the wheel. In order to strengthen still further this fastening, rings *e* may be placed inside the bent paddle ends (Fig. 207), and a ring *f* may also be shrunk on around the outer ends of the fixed paddles (Fig. 207), care being taken to give the paddle ends such a form, either convex or concave, that the wheel ring during the shrinking on obtains a corresponding form and is thus more securely held in position.

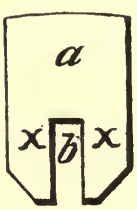


Fig. 208.

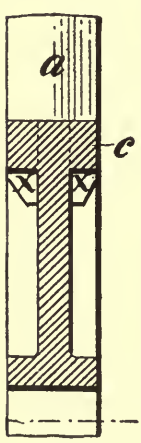


Fig. 208A.

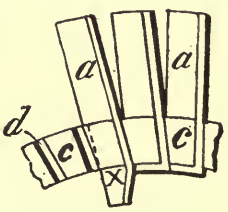


Fig. 209.

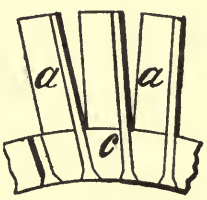


Fig. 209A.

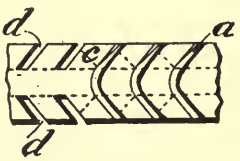


Fig. 209B.

Instead of bending over the inner ends of the paddle blanks (Fig. 209), they may be fastened to the wheel body by riveting their ends as illustrated in Fig. 209A.

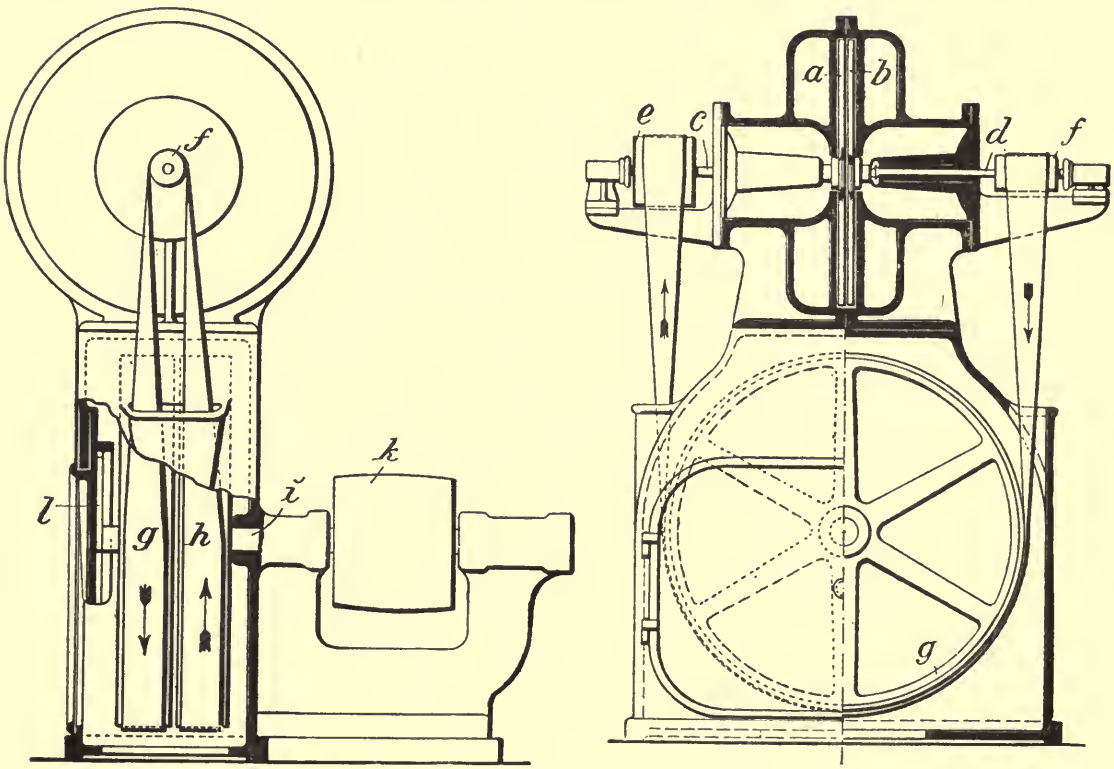
His next improvement consists of a method for coupling the two wheels to one shaft by a speed reducing gear. He prefers belt gearing for the purpose, and the method is thus shown and described.

The inventor is aware of the fact that a single belt curved round pulleys has been used in this way before. He claims as new the fact that he delivers the power to the

belt by two pulleys, one on each turbine wheel shaft to a fast pulley on the shaft to be driven, and a loose pulley which acts as guide pulley, and is made movable in order to tighten the belt when necessary.

There can be no doubt that for small powers a belt, if well chosen of right material and proper breadth and thickness, can be expected to give good results in the double duty of reducing speed and transmitting the power of two wheels to one shaft.

In steam turbines the power is usually transmitted from the turbine shaft to the main shaft by means of toothed gearing; the gear teeth, owing to their great velocity, frequently break, rendering the motor unserviceable. For this reason the use of a belt for the power transmission is preferable, more especially as a broken belt can be replaced or repaired without appreciable expense or loss of time.



End View. Longitudinal Section.

FIG. 210.—Seger Turbine.

Referring to the illustrations—

Fig. 210 illustrates a longitudinal section of a compound steam turbine provided with the arrangement referred to, and an end view (partially in section) of the same.

a and *b* are the two turbine wheels which by the steam passing through are caused to revolve in opposite directions, as indicated by the arrows in the Figures. Their shafts *c* and *d* (which are horizontal in the turbine illustrated) point in opposite directions, and on each of them is secured a belt pulley *e*, *f*, of a diameter proportioned to the respective velocities of the two turbine wheels so as to give the same velocity to the peripheries of the two pulleys. Below, above, or at one side of these wheels there are placed two pulleys *g h*, one of these (*h* in the drawing) being fixed to a shaft *i* rotating in fixed bearings and provided with a driving pulley *k*. The other pulley *g*, on the other hand, is secured to a short shaft, which may also be journalled in a fixed bearing, though

the better plan is to journal it, as shown in the drawing, in a sliding carriage *l* arranged in the engine frame in such a manner that it can be moved towards or from the axis of the turbine wheels, thus retaining an even tension in the belt wrapped around the pulley *g*. When the latter is located below the turbine wheels so as to be suspended in the belt, as in the drawing, it will maintain an even tension on the belt by virtue of its own weight; if this is not the case, the tension may be kept uniform by means of a spring or weight tending to force the carriage together with the pulley *g* outward from the axis of the turbine wheels. Around the 4 pulleys *e*, *f*, *g*, and *h* is wrapped a belt passing from the pulley *f* around the reverse side of the pulley *g*, then over the pulley *e* on the same side as over *f*, and finally reaching the pulley *f* on passing around the reverse side of the pulley *h*. By this method the power is transmitted from the turbine wheels, rotating in opposite directions, to the shaft *i* and the driving pulley attached to it.

We may next notice the patents of Mr. Richard Schulz, of 1900, No. 21,472. His ambition is to replace marine engines by turbines and balance the axial thrusts. He starts out in November 1900 to tell us what Parsons had long ago demonstrated on land and sea. He says:—

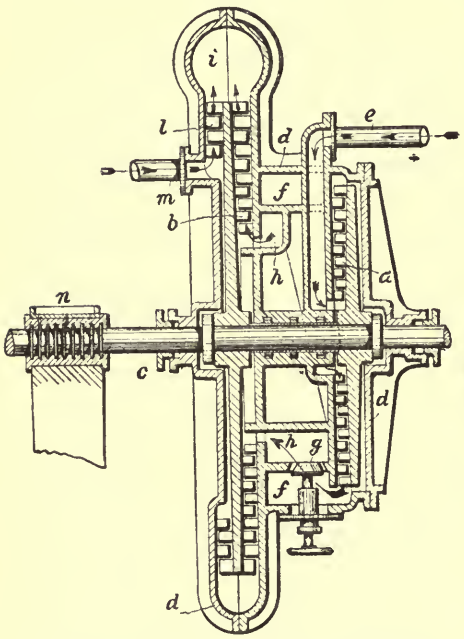


FIG. 211.—Schulz Radial Turbine, with Reversing Turbine.

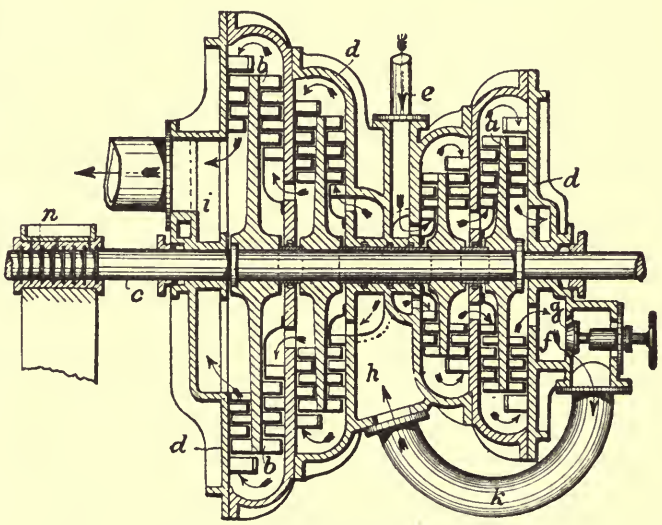


FIG. 212.—Schulz Radial Turbine.

"Steam turbines, on account of the regular force which they exert upon a shaft, possess many advantages over steam engines with reciprocating pistons.

There are, however, considerable drawbacks in the ordinary construction of steam turbines in comparison with steam engines, on account of the very considerable axial pressure which is imparted to the shaft under certain circumstances.

In order to avoid this axial pressure, two similar steam turbines have been mounted on the same shaft symmetrically side by side, to exert an axial pressure in opposite directions. Such double steam turbines, however, require much room and are of great weight, and cannot work so economically as only a single turbine. Counter pressure discs firmly connected with the shaft have been employed in the case of steam turbines, with turbine coils arranged axially one behind the other, and also in steam turbines with radially arranged turbine coils in which the counter pressure discs are encountered by the steam in an opposite direction to the turbine body, thus preventing an axial pressure which tends to displace the turbine body in the casing.

In addition to the fact that the arrangement of such discs is a drawback because they take up space, there is also the impossibility of rendering these discs steam-tight on the periphery, and consequently a considerable loss of steam is unavoidable.

These drawbacks are avoided by the arrangement of steam turbines which form the object of the present invention, and which, in the first place, are intended for replacing marine engines. The arrangement is characterised by a smaller turbine working under a higher steam pressure, and a larger turbine working under lower steam pressure being arranged on one or more shafts in such a way that their axial pressures take place in opposite directions, in order to entirely remove the resultant effective axial pressure falling on the bearings, or to prevent it at least to a sufficient extent."

In one form of construction of steam turbines (Fig. 213) turbine coils are axially arranged, having a smaller diameter in the turbine body which works at a higher pressure, and a greater diameter in the turbine body which works at a lower pressure. The two turbine bodies *a* and *b* are mounted on the same shaft *c*, preferably in a rigid casing *d*.

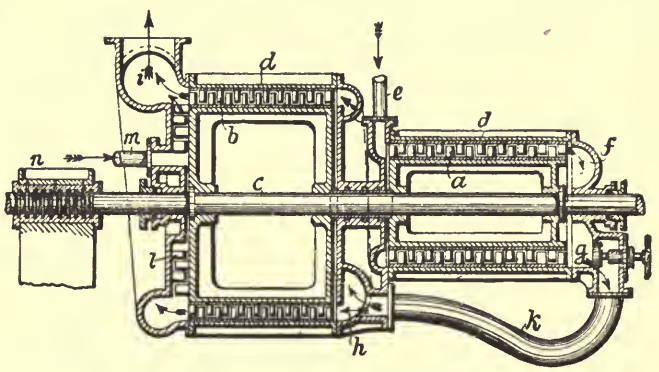


FIG. 213.—Schulz Axial Turbine, with Reversing Turbine.

The steam enters through a nozzle *e* arranged about the centre of the length of the casing, and impinges against the vanes of the smaller turbine *a*, and quits this latter through an overflow passage *f*, passing into the inlet passage *h* of the larger turbine *b* through a pipe *k* provided with a regulating valve *g*. This larger turbine is traversed with steam of lower pressure in a direction opposite to that in which it flows through the smaller turbine; the steam finally quitting the compound turbine through a discharge pipe *i*.

The direction of the vanes in both turbines, and the direction of the guide passages in the casing, must of course be so selected as to correspond with the desired direction of rotation of the shaft.

The regulating valve *g* in the overflow passage may, in case of need, retain a portion of the steam escaping from the smaller turbine, and produce a higher steam pressure behind the said turbine and diminish the pressure of the steam flowing towards the larger turbine through the free opening of the valve. By adjusting this regulating valve, the resultant axial pressure on the shaft which falls on the bearings may be altered within certain limits. By means of guide vanes or coils *l* provided on one end wall of the casing, which is also formed into an exit passage, an additional turbine is formed for backing, which is necessary to enable ships to be manœuvred. In this case fresh steam flows through an inlet pipe *m* on to the smallest of these concentric turbine coils, and passes out of the largest turbine coil into the exit passage in the casing end. When the movement is in a forward direction, this backing turbine runs without deleterious action in the diminished pressure of the exhaust or discharge steam.

In another somewhat similar form of construction of steam turbine provided with high pressure and low pressure turbine coils, such as shown in Fig. 214, these coils increase in size in the direction of the flow of the steam. A similar arrangement may also be adopted if the turbine coils arranged one behind the other become smaller in

diameter instead of increasing in diameter in the direction of the flow of steam, or if a portion of the turbine coils have approximately a constant diameter, whilst the rest increase or decrease in diameter as required. Schulz's radial flow turbine is shown in Fig. 211, in which the high pressure turbine *a* and the low pressure turbine *b* are provided with discs of different sizes fixed on the shaft *c* of the engine in the casing *d*, which discs carry radially arranged turbine vanes. These two discs carry on the sides facing one another a larger number of turbine vanes, for the guidance of which guide passages or coils are provided on the opposite walls of the casing *d*. In this case also the steam enters through a pipe *e* lying approximately at the middle of the casing, and quits the high pressure turbine *a* through an overflow pipe *f* provided with a regulating valve *g*, and passes through an inlet passage *h* into the low pressure turbine *b*, and finally quits the combined turbine through an exit passage *i* at the outside. In the example shown the steam passes into the high pressure turbine *a* and also into the low pressure turbine *b* at the smaller of the concentric turbine coils, and passes out at the largest of the turbine coils. The disc of the low pressure turbine *b* carries on its reverse side, near its periphery, suitable vanes for a backing or reversing turbine *l*. The fresh steam enters through an inlet pipe *m* here also at the smallest of the concentric turbine coils, and escapes into the exit passage *i* from the largest of the turbine coils.

In another arrangement shown in Fig. 212 both the high and low pressure turbines *a* and *b* are provided with several discs fixed on the driving shaft *c* in the casing *d*, which discs carry turbine vanes on both sides. The steam passes through the inlet pipe *e* lying approximately at the middle of the length of the casing, first into the smallest disc of the high pressure turbine *a*, and quits the latter at the larger disc in order to pass through an overflow passage *f* provided with a regulating valve or device *g*, and through a pipe *k* into the inlet passage *h* to the low pressure turbine *b*, finally quitting the latter at the largest disc through an exit passage *i*.

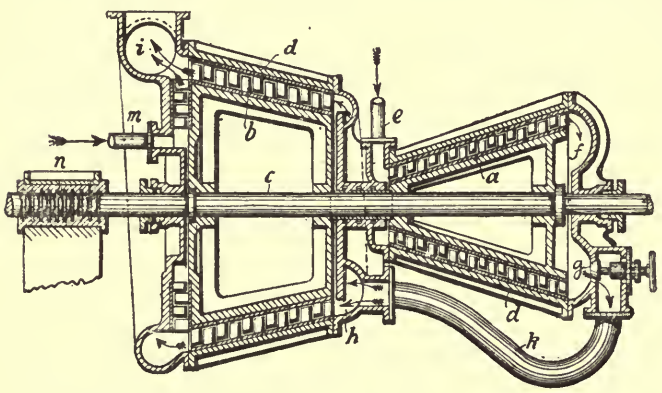


FIG. 214.—Schulz Axial Turbine, with Reversing Turbine.

If it be necessary to arrange this form as a backing or reversing turbine, this may be easily done in the same way as in the examples previously described.

Of course the compound turbine may be composed of more than two turbine bodies. According to the foregoing, the arrangement may easily be such that the resulting axial pressure which is imparted to the shaft *c* and the bearing *n* may be brought to the desired minimum. The same result is also easily attained by the foregoing if the separate turbine bodies work in separate casings, and if they are also distributed over several shafts. Arrangements for regulating turbines are given as follows:—

The lever which works the rod 17 (Fig. 215) may also be connected with a powerful governor in case the adjustment of the annular slides is to take place automatically. Further, in the normal position of the revoluble slide 8 (Fig. 216), in which position all the openings are exposed, as well as in any other position, the section of passage of each separate guide blade ring 7 may also be suitably diminished or increased by turning the latter itself, as the edges of the respective guide vane openings pass more or less beneath the imperforate portions of the respective slide 8. The pressure of the steam

on entering a fresh turbine compartment may thus be reduced from the first to a given extent.

In the case of a steam supply which only extends to a portion of the circuit in which the guide vanes and the steam passages formed by them are distributed, for instance, by a distribution of the inlet apertures in the guide blade ring 7 in the manner shown in Fig. 215, the annular slide 8 has the form shown in Fig. 215. In a normal position (greatest effectiveness) the openings of the annular slide 8 lie over the openings in the guide blade ring 7 which forms its seat, but when displaced relatively to the guide blade ring 7 the openings of the latter are successively covered by the imperforate portions of the slide.

In the radial turbine shown in Fig. 216 the arrangement of the annular slide 8 and its seat is such that an exactly similar action of steam distribution is obtained.

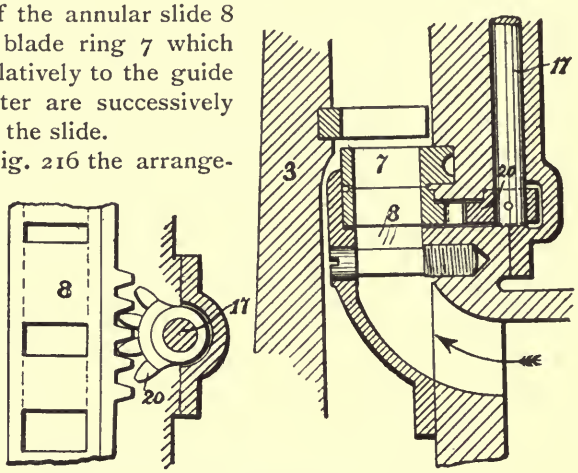


FIG. 215.—Revoluble Slide of Schulz Turbine.

15 here indicates the guide vanes which are fixed on the walls of the casing, and 2 the pressure vanes of the turbine bodies mounted on a shaft passing through the various compartments of the casing. The relative adjustment of the annular slides 8 is here effected by means of shaft 17, which is passed upwards through the walls of the turbine casing and carries toothed segments 20, which engage with teeth on the annular slides 8 (Fig. 215).

The steam enters through a pipe socket at one end of the casing against the turbine body of smallest diameter, and escapes from the casing through a pipe socket in the cover.

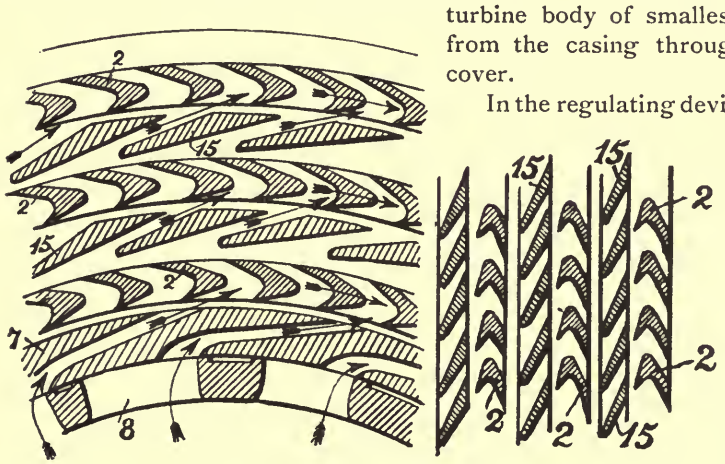


FIG. 216.—Schulz Turbine Blades and Guide Blades.

In the regulating device hereinbefore described, in which, in addition to the regulation of the steam consumption to correspond with the effectiveness of the work, a second regulation is possible by the adjustment of the guide blade ring 7 of the annular slide 8, a subsequent improvement of the calculated and effective steam sections is possible also for the greatest resultant power, which is very valuable for compound steam turbines. By this means it is possible to diminish or enlarge the inflow apertures for each separate turbine body separately until the correct pressure forming the basis of the calculation on the entrance of the steam into the turbine bodies is actually attained, which may be ascertained by means of a manometer.

The arguments set forth for the necessity for the improvements above described the patentee sets forth as follow:—

"Steam turbines having several rings of blades, whether axially or radially arranged, work with comparatively great economy only so long as they develop their greatest effectiveness when the sections of the steam passages are calculated for this greatest effectiveness, so that the steam enters the first ring of the turbine at as high a pressure as possible, and only gradually loses pressure in its passage through the others. If, however, temporarily a less amount of power is required from the same turbine with the same size or section of the steam passages, and a suitably diminished quantity of steam is conveyed to the turbine by the action of a throttle valve, the steam on its entrance into the first ring of the turbine has already lost too much pressure, and therefore is utilised uneconomically. In order to avoid this drawback, by the present invention the sections of the steam passages are regulated to correspond with the desired effectiveness or work required. For this object, in front of all the rings of guide blades or vanes, or in front of some of them, annular slides with openings or apertures are arranged in such a way that with a given position of the annular slide a larger or smaller number of openings to the steam passages which are formed by the guide vanes and pressure vanes may be closed.

In a turbine which replaces marine engines it is more particularly advantageous

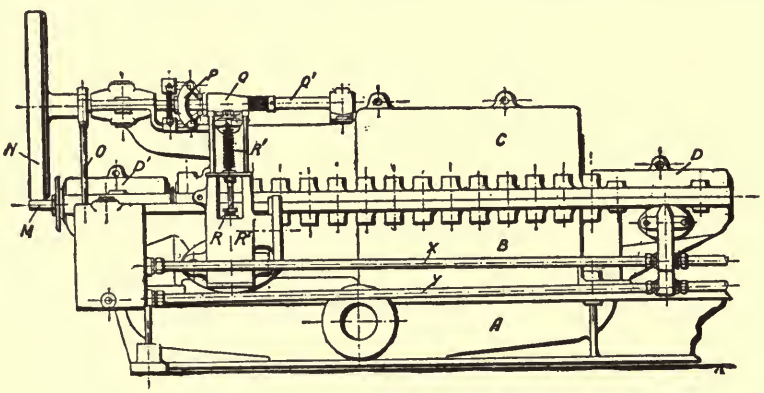


FIG. 217.—Schulz Turbine Complete.

to adopt a diameter for the turbine bodies forming the combined turbine, by which still an effective pressure in the direction of the shaft results during the travel of the turbine in order to strain the bearings *n* of the shaft as little as possible. This effective pressure is directed backwards when the turbine is so used, and must be so selected that it approximately equals the impulse given to the propeller in a forward direction. It is advisable to attach manometers at suitable places on the high and low pressure turbines *a* and *b*, in order to permit of the steam pressure being ascertained at any time. By means of the valves or regulating devices *g* arranged in the overflow passage *f* between the high and low pressure turbines, small alterations in the pressure of the steam behind the high pressure turbine body *a* and in front of the low pressure turbine body *b* may be easily effected by observing the steam pressures in the manometer."

He then, in 1901, proceeds to show how to regulate the power of turbines, by regulating the sections of the steam jet, by a method about fifty years old, as applied to water turbines of impulse type, *i.e.* by sliding disc or ring covering more or less of the steam ports or passages.

In the construction shown in Figs. 218 and 219 a shaft 1 carries five turbine bodies in series 3 provided on the periphery with pressure vanes or blades such as 2 (Fig. 216) is revolubly mounted in the enclosing covers 4 and 5 of the casing 6. The casing 6 is at

one end wider than at the other. The cover 4 at the smaller end has a pipe socket for the admission of the steam, the other cover 5 has a pipe socket for the escape of the steam.

The separate turbine bodies 3 (of which there are five in the present example) lie in separate compartments of the casing, which are formed by partitions 7 tight jointed on the shaft 1. A larger or smaller number of passages which are contained in these partitions 7 may be closed by means of ring slides 8 lying in front of them.

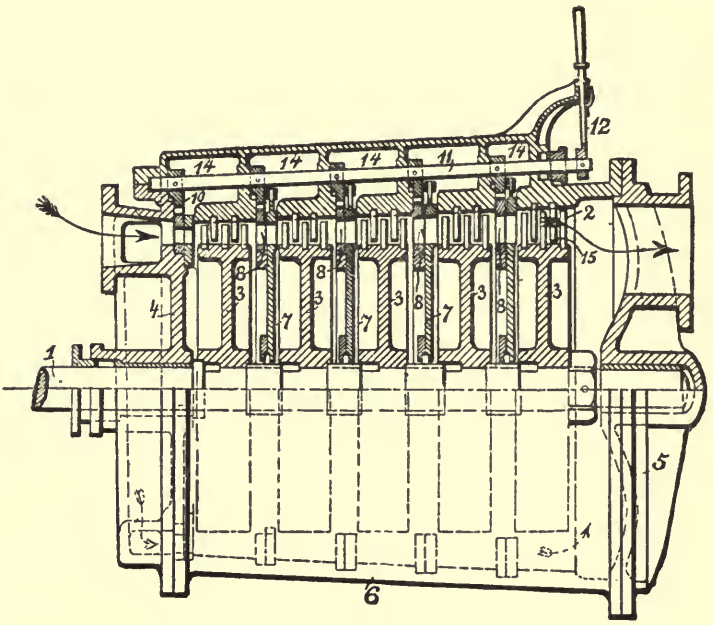


FIG. 218.—Turbine, with Regulator.

These ring slides are provided on the upper part of their periphery with teeth 9, in which toothed segments 10 mounted on shafts 11 engage, which segments may be adjusted by means of a setting lever 12 on a curved scale or setting arc 13. The shaft 11 passes through the compartments 14, which are formed by upper extensions of the casing 6, and are tight washered at one end by means of stuffing boxes.

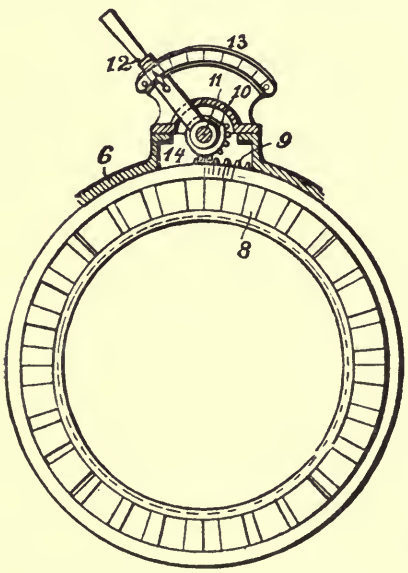


FIG. 219.—Turbine Regulator.

The guide vanes 15 forming the said passages are fixed on the inner periphery of the casing 6. Steam flows between these guide vanes against the pressure vanes 2 of the separate turbine bodies.

Rods passing through stuffing boxes on the side walls of the compartments 14 so as to form a tight joint, and provided at the outer end with a screw thread and nut, engage above on the partitions 7, in order to enable these partitions to be also adjusted relatively to one another by the adjustment of the nuts.

The steam passes through the pipe at the end 4 into the turbines, flows through these latter in the direction of the arrow, gradually losing its force, and escapes through the pipe in the end or cover 5. Before the steam can enter the steam passages of the first ring of guide vanes it passes through the openings of one of the annular slides 8 lying in front of the steam passages. The walls or plates 7 serving as seats for the ring slides have in the example shown 28 admission openings of equal section and equally distributed round the circle. The ring slide (Fig. 219) has also 28 openings coinciding in each quarter of the ring, which are of different width and at different distances

apart. The distances are so calculated that the ring slide in the one end position, which is rendered clear by the projection of a quarter in section in Fig. 220 (upper Fig.), exposes all the inlet apertures of the partition 7. When the annular disc is displaced in the direction of the arrow (Fig. 220), the opening lying farthest to the left of the wall is first closed, and then the next opening, and so on simultaneously in all four quarters, until the position shown in Fig. 220 (lower Fig.) is reached, in which only one opening of the wall 7 remains uncovered in each quarter. How many openings are closed by the ring slides may be ascertained on the scale, which is divided into 7 parts on the setting arc 13. By moving the setting lever 12 to the extent of one section of the scale, all the annular discs 8 are so turned that 4 of the 28 openings of each wall 7 are closed.

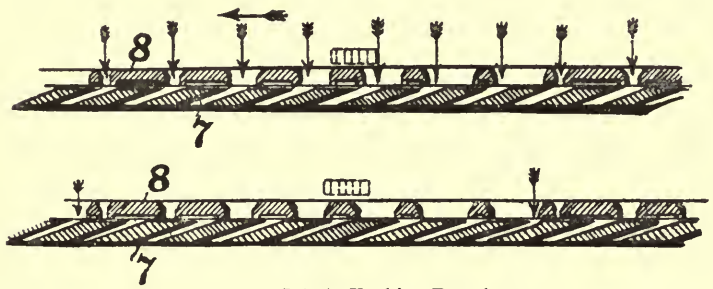


FIG. 220.—Schulz Turbine Regulator.

Another German engineer is in the field, Johann Stumpf, Berlin, with a patent 18,952, 1902, for a compound turbine like Rateau's—essentially a number of De Laval wheels in series. Rateau and Laval turbines are axial flow machines.

Stumpf affects the inward flow, and he shows us how to cut off part of the turbine when on small load by stop valves or slide valves.

This invention relates to improvements in steam or gas turbines, and more especially to a turbine in which the steam is expanded and the pressure of the same is transferred into working energy.

The object of this invention is to so construct a turbine that the number of revolutions of the turbine can be changed at will.

The construction is shown in the illustrations, in which—

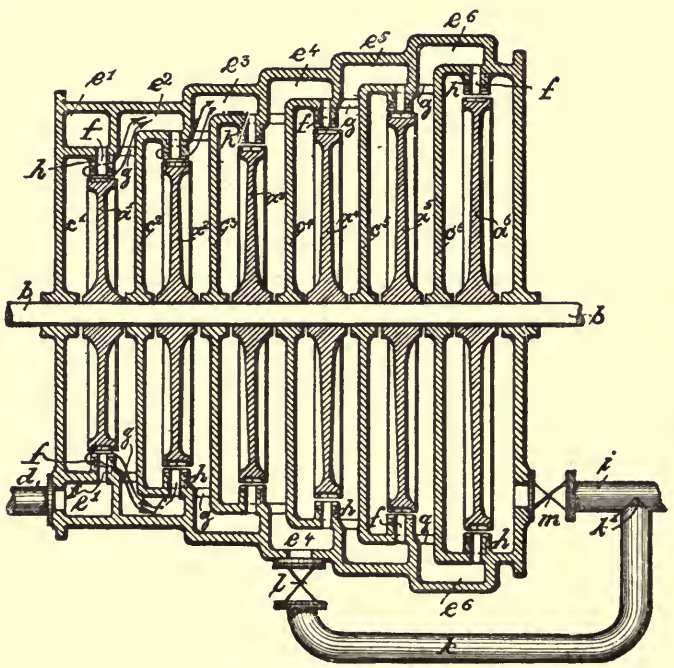


FIG. 221.—Stumpf Turbine.

Fig. 221 is a vertical section of a turbine, constructed according to this invention, and

Fig. 222 is a vertical section of a modification of the turbine shown in Fig. 163.

The invention consists in so constructing the turbine that one or several turbine wheels can be cut out in such a manner that a greater or smaller number of turbine wheel rims come into action for effecting the transformation of the steam pressure into energy.

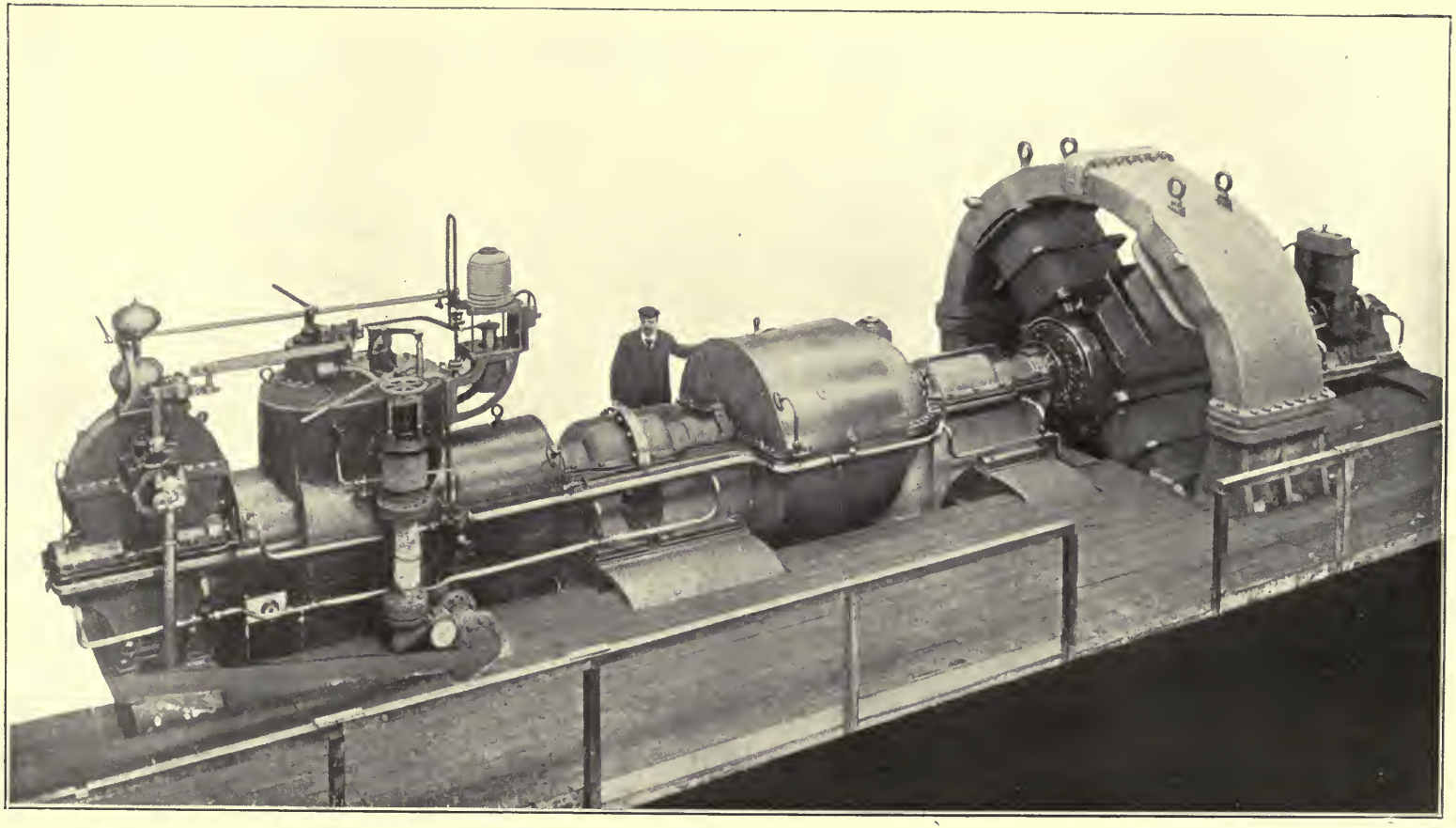


PLATE VII.—PARSONS' TURBO ALTERNATOR.

The invention may be carried out in different ways according to the construction of the turbine, and especially according to the circumstance whether the pressure of the steam is utilised in the turbine in a series of turbine wheel rims step by step, or whether the steam expands in the first admission nozzle up to the exhaust or condenser pressure, so that the steam works only by its velocity.

In turbines of the first mentioned type, one or several turbine wheels can simply be cut out by leading the steam before it comes to the last turbine wheel or turbine wheels to the condenser or into the atmosphere. In turbines of the latter type the object may be attained in a very simple manner by arranging the turbine wheels or the leading vane rims, so that the same can be shifted sideways in the direction of the axis of the wheels, so that the steam, after having worked in a number of turbine wheels, flows immediately into the condenser or into the atmosphere.

Fig. 221 relates to the first type of turbines, that is, to a turbine in which the steam expands in each nozzle step by step. a^1, a^2, a^3, a^4, a^5, a^6 are turbine wheels fixed to the common shaft b. Each turbine wheel rotates in a chamber c^1, c^2, c^3, c^4, c^5, c^6 of the common turbine casing. The live steam enters at d an annular channel e^1, and streams through nozzles f upon the first turbine wheel a^1. In the example shown in the drawing, turbine wheels with double buckets are shown, so that the steam enters on both sides of the turbine wheel into the chamber c^1. The steam leaving the turbine wheel on the right-hand side of the same (as shown in the Figure) immediately flows through ports g to the second annular channel e^2. The steam turbine leaving the first turbine wheel on the left-hand side flows through passages passing through the ring h containing the nozzles to the right-hand side and flows into the second annular channel e^2, the direction of flow of the steam being shown by the arrows in the drawing. From the annular channel e^2 the steam then impinges upon the second turbine wheel a^2, and so on. The steam leaves the last chamber c^6 through the pipe i, which leads to the condenser.

In the Figure a special pipe k for the annular channel e^4 is shown, which pipe is connected at k^1 to the condenser pipe i. The pipe k has a valve l, and the pipe i has a valve m between the turbine casing and the spot k^1 where the pipe k is connected to the pipe i. If the valve m is closed and the valve l is opened, the steam flows from the annular channel e^4, that is to say, after having worked in the first three turbine wheels a^1, a^2, a^3, to the condenser. In the Figure the annular channel e^4 is only shown connected to the pipe i, but it will be understood that this connection may be also provided for all the other annular channels, so that each of the channels can be connected directly to the condenser.

If, as is the case in this construction, the steam has to be passed into a smaller number of turbine wheels than the total number contained in the turbine, owing to a reduction in the pressure of the steam, it is natural that, corresponding to the reduction of the pressure of the steam in each nozzle system, a greater cross section of the admission nozzles for the single wheels must be obtained. This can be attained, for instance, by cutting out a number of nozzles and increasing the area of those still open, and this may be accomplished in any suitable manner, for instance, by means of an annular slide or the like.

In Fig. 222 the invention is shown as applied to a turbine, in which the steam expands immediately to the pressure of the condenser or exhaust, so that the whole pressure is immediately transferred into streaming velocity. In the example a steam turbine is shown in which three turbine wheels b, b^1, b^2 are arranged upon a common shaft a. The turbine wheel b situated in the middle is provided with a turbine wheel rim c, in which are arranged the well-known double buckets after the manner of the Pelton wheel system. The two outer wheels b^1 and b^2 have wheel rims d^1 and d^2, in each of which rims are provided two rows of single buckets of, for instance, a U-formed cross section. The steam flows into the turbine through an annular channel e, and

streams through nozzles f to the double buckets of the wheel b. In order to lead the steam from these buckets to the additional turbine wheel rims, four leading vane rims g^1 g^2 and h^1 h^2 are provided. The steam jets entering through the nozzles f and impinging upon the rim c of the turbine wheel b are divided by means of the double buckets into two parts. These parts stream through the buckets and enter the leading vane rims g^1 g^2, in which the streaming direction is reversed, so that they enter again in the right direction into the first rows of buckets d^1 and d^2. After streaming through these buckets the streaming direction is again reversed in the leading vane rims h^1 and h^2, so that the steam is led into the last row of buckets of the turbine wheels b^1 and b^2, from where it enters into the turbine casing in order to exhaust or to be condensed.

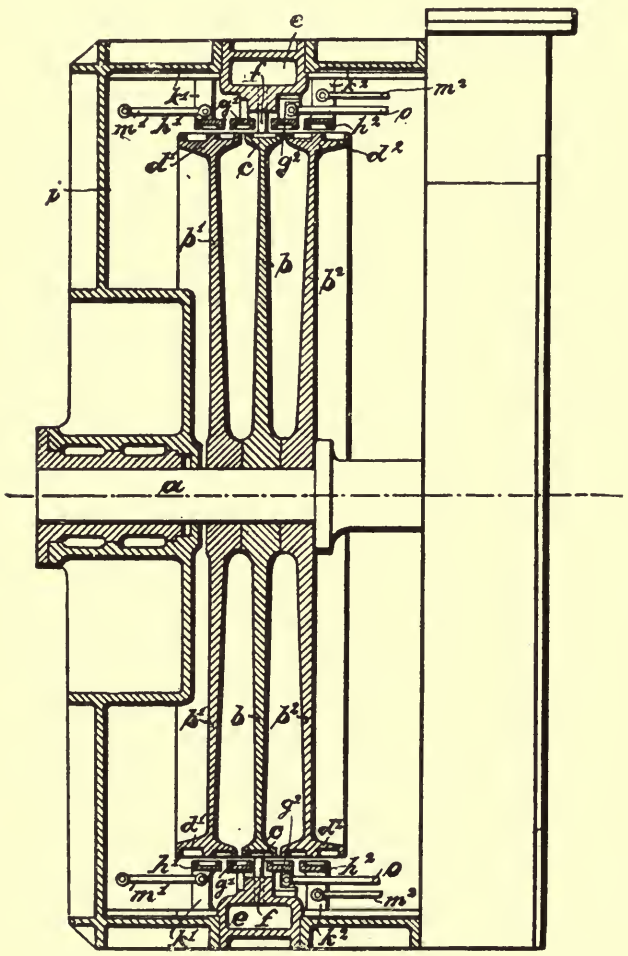


FIG. 222.—Stumpf Turbine.

Several or all of the leading vane rims g and h are movable sideways. In the Figure the leading vane rims h^1 and h^2 are provided with guide pieces k^1 and k^2, by means of which they can be moved sideways in the turbine casing i. The devices for moving the rims may be of any convenient form. In the Figure, rods m^1 and m^2 are shown, the number of which rods is equally divided up over the whole circumference of the respective vane rims, means being provided for moving all the rods m^1 and m^2 simultaneously by means of a common lever mechanism. Also for the leading vane rim g^2 guide rods o are shown, in order to make clear that the outer as well as the inner vane rims are movable, either singly or both together.

If, for instance, the number of revolutions for the number of turbine wheels and leading bucket rims shown in the drawing, and for a diameter of the turbine wheels of 3 metres, amounts to 500, and the outer leading vane rims h^1 and h^2 are moved sideways, the number of revolutions increases up to 1000 for nearly the same steam consumption. If, now, the following leading vane rims are moved sideways so that the steam streams out freely from the first turbine wheel, the number of revolutions should increase up to 2000 in order to obtain about the same steam consumption.

In this example an augmentation of the number of revolutions from 500 to 1000, and from 1000 to 2000 is attained. Naturally, it is also possible to create intermediate steps. It will be clear that the arrangement shown and described cannot only be used in turbines provided with double buckets after the Pelton wheel system, but that it may also be used for turbines in which the steam jet is not divided up into two parts.

As the Curtis turbine is being pushed forward in this country a few extracts from some of his patents are of interest. It may be said at once that although we have three voluminous specifications before us it is very difficult to discover much new to select from the descriptions and illustrations. From one specification, No. 19,246, 1896, we may take a sample or two. This specification has fourteen pages closely printed, and elaborately tells us all over again what Parsons and De Laval had told us ten years before, and Wilson and Pilbrow thirty years before that. As an example of an American specification covering some small detail improvements, it is difficult to surpass it. There are seven claims and nine sheets of drawings. A few illustrations from the nine sheets with their explanations are given thus :—

In Fig. 223 is represented a turbine having four sets of movable vanes D, D^1, D^2, D^3, and three stationary intermediate passages E, E^1, E^2, constructed and operating on this principle. In this case, and assuming that the apparatus is designed to work between the initial and terminal pressures before taken for illustration, *i.e.* a boiler pressure of 150 lbs. and an exhaust pressure of 2 lbs. per square inch (absolute), the delivery nozzle G will be proportioned in the manner described so as

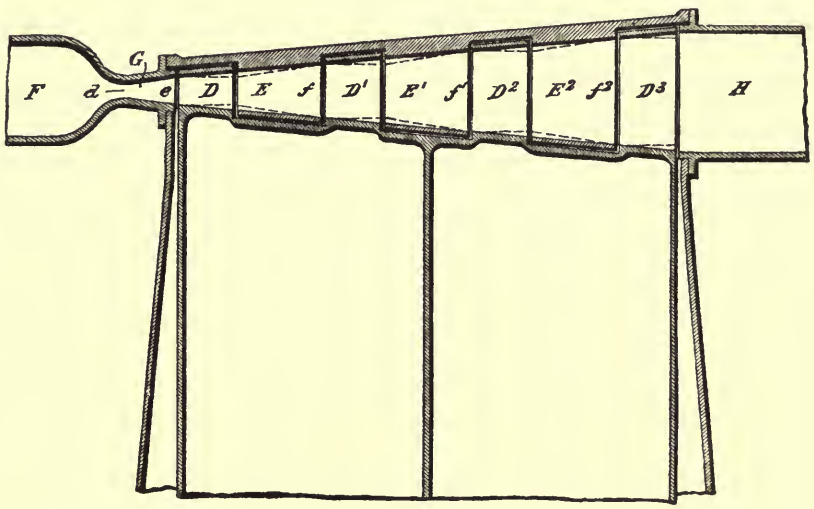


FIG. 223.—Curtis Turbine.

to expand the steam down to a given pressure, say 10 lbs. at its discharging end, the remaining pressure above the pressure of the exhaust being that assumed to maintain the flow in the working passages. If the steam jet enters the exhaust with 20 per cent. of the original velocity, each of the four sets of movable vanes will abstract 20 per cent. of the velocity, and will in their order absorb respectively 36, 28, 20, and 12 per cent. of the energy represented by the original velocity, while 4 per cent. of the energy will pass off as residual velocity in the exhaust. The decline in pressure in the working passages will be such that at the discharge ends f, f^1, and f^2 of the stationary intermediate passages the pressures will be less and less, declining at a certain rate which should be found by experiment in each particular type and size of machine, and owing to both reduced velocity and to increased volume of the fluid jet from reduced density the cross sectional areas at the points f, f^1, f^2 will be correspondingly increased over the cross sectional area of the discharge end e of the nozzle, and over each other successively. The expansion in the working passages is shown in full lines as taking place in both the movable and stationary passages, while by the dotted lines this expansion is shown as taking place wholly in the stationary passages. The former plan is preferable.

In Fig. 224 is shown an apparatus designed to have the same proportions of the expanding nozzle and the working passages and to operate under the same conditions as the apparatus of Fig. 223. It has the peculiarity in construction of having the different sets of movable vanes mounted upon separate wheels. It also differs in the construction of the stationary intermediate passages which will be presently described.

Besides the forms of apparatus which have been described, many features of the invention are involved in the construction and operation of such an apparatus as that shown in Fig. 225, in which only movable vanes are employed in the working part of the apparatus, the sets of vanes being alternately mounted upon oppositely rotating discs, and delivering the fluid jet from one set of movable vanes directly to another set without the interposition of stationary passages. It will be understood that the movable vanes have all the characteristics of the movable vanes already described, but that those on one disc are set or curved oppositely to those on the other disc. In the apparatus illustrated in Fig. 226 the expanding nozzle is designed to convert all or the larger portion of the useful pressure into *vis viva*, although the overlapping of the operations of converting pressure into *vis viva* and *vis viva* into mechanical power can likewise be employed in an apparatus of this kind.

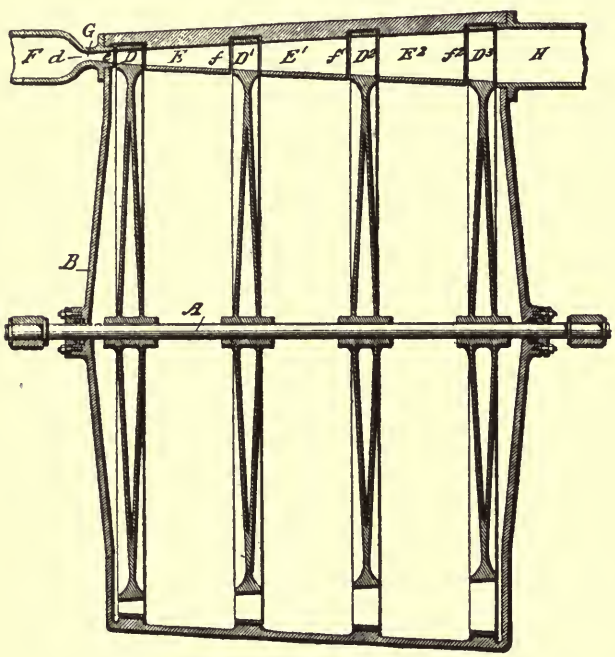


Fig. 224.—Curtis Turbine.

It will be observed that in all forms of the apparatus the steam or other elastic fluid is delivered to the working passages practically in the form of a solid stream or "jet," whose cross sectional area is large compared with its perimeter, and that this jet acts at one time only on a small section of the vanes of a circular range. This arrangement has a great practical advantage over one wherein the fluid is delivered in the form of an annular film simultaneously to all the vanes of a complete circular range. In the latter arrangement, in order to reduce the total cross section of the passage to that required and still have the proper velocity, the depth of the vanes has to be exceedingly small, and the surface exposed to the flowing fluid is very large compared with the cross sectional area of the passages. The result is great loss by surface friction and churning action, and in addition the clearance space is so extended that it becomes large in proportion to the area of the passages.

Whether the turbines are now made on these principles and designs we cannot ascertain, but large turbines of some kind or another, the leading feature of them being that they have vertical shafts, are being placed on the market.

On the whole, it will be seen from all this that very little has been contributed to the improvement of steam turbines by any specification other than De Laval and Parsons. Arrangements for governing and regulating, placing turbines side by side, or end to end, or one inside the other, forming vanes, nozzles, and jets, constructing

wheels for high velocity, yielding bearings and flexible shafts are subjects upon which inventors and patent agents have been, and no doubt will continue, ringing the changes for some time to come.

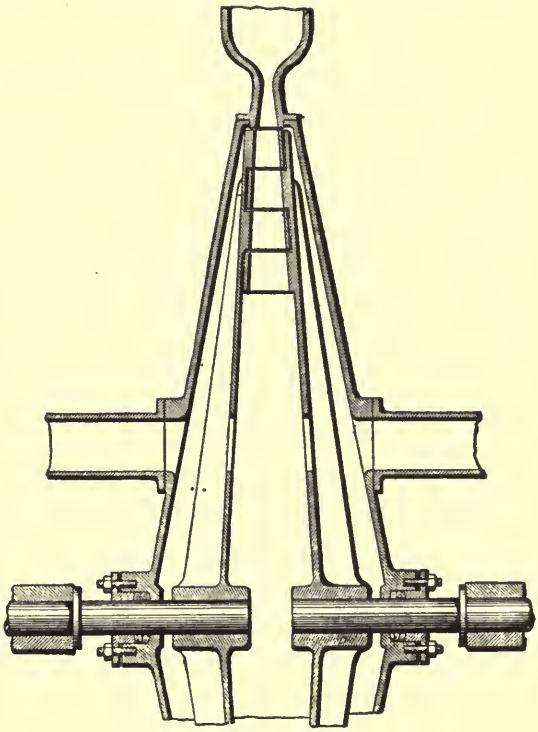


FIG. 225.—Curtis Turbine.

STEAM TURBINE TESTS

It is necessary here to give some results of steam turbine tests to form some idea of their practical performances. Some tests we have briefly referred to already, so that we may select only a few typical ones. We shall have occasion in future volumes to refer to their special performances in connection with the work to which they are applied —marine propulsion, pumps, fans, and dynamos. Parsons and De Laval turbines are the only ones at this date upon which reliable independent tests have been reported, so that we are unable to give results for the later turbines.

Up till 1890 Parsons turbines had not been made condensing, but were chiefly of small sizes for electric lighting, as we have seen from the patents.

The next step in advance was the construction of an experimental steam turbine of 200 horse-power in 1892. It was coupled

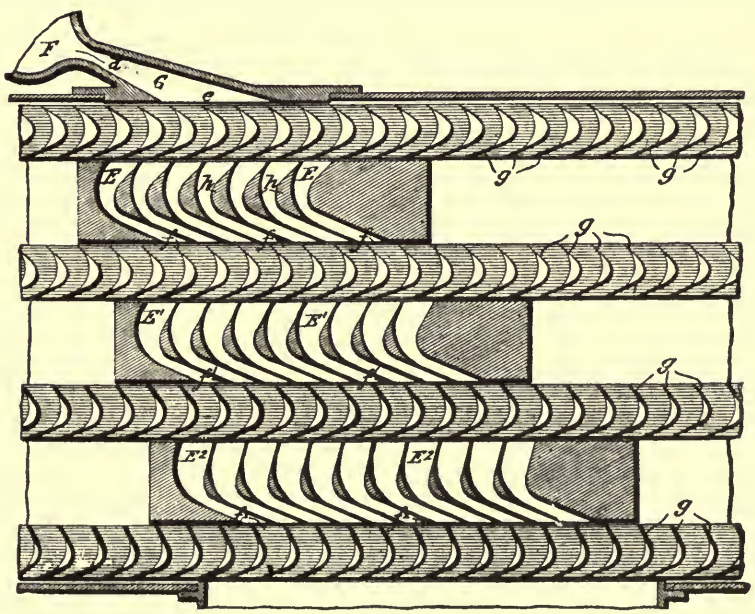


FIG. 226.—Fixed and Movable Blades of Curtis Turbine.

to a 100-kilowatt alternator, and supplied with moderately superheated steam at 100 lbs.

pressure per square inch. When tested by Professor J. A. Ewing, F.R.S., it was found to consume 27 lbs. steam per kilowatt-hour, thus rivalling the performances of the best compound condensing reciprocating engines. This result placed the steam turbine amongst the most economical means of obtaining electrical energy from steam, and led to its adoption in the lighting stations of Newcastle, Scarborough, Cambridge, and other places.

About two years later considerable alterations of design and workmanship were introduced; a single flow type of parallel flow turbine having been adopted instead of the original double flow with right- and left-handed turbines on each side of the steam inlet, the second set of turbines were replaced by rotating steam balance pistons, and the steam passed in one direction parallel to the shaft. This alteration materially improved the economy and reduced the amount of skilled labour required. The form and construction of the vanes or blades was perfected and strengthened, and many minor improvements conducive to economy were made, so that even in the smaller sizes a fair degree of efficiency has been obtained, as is instanced by the result of 28.8 lbs. of steam per kilowatt-hour, or about 17 lbs. per indicated horse-power, for a 24-kilowatt steam turbine plant without superheat.

The following tables will give the general results of reliable tests on Parsons' turbines:—

TABLE XVII.—TEST OF 24-KILOWATT TURBO DYNAMO FOR MESSRS. SPILLERS & BAKERS, NEWCASTLE-ON-TYNE.

Pressure of Steam above Atmosphere at Stop Valve.	Superheat at Stop Valve.	Vacuum in the Turbine Cylinder. Bar. = 30″.	Revolutions per Minute.	Load.	Steam used.	
Lbs. per Square Inch.	F.°	Inches of Mercury.		Kilowatts.	Lbs. per Hour.	Lbs. per Kw.-hour.
80	0	28.8	4990	24.7	712	28.8
77	0	29.0	4630	11.8	400	33.9
74	0	29.1	4570	5.15	235	45.6
78	0	26.0	4900	23.8	798	33.5
79	0	0	4780	19.7	1350	68.5

The increase of efficiency with better vacuum is well shown here. With 28.8 inches vacuum this is about 4.6 lbs., or 16 per cent., better than with 26 inches vacuum.

A 50-kilowatt steam turbine alternator for the Corporation of Blackpool showed a consumption of 28 lbs. per kilowatt-hour at full load without superheat.

TABLE XVIII.—50-KILOWATT STEAM ALTERNATOR FOR THE BLACKPOOL CORPORATION.

Pressure of Steam above Atmosphere at Stop Valve.	Superheat at Stop Valve.	Vacuum in the Turbine Cylinder. Bar. = 30″.	Revolutions per Minute.	Load.	Steam used.	
Lbs. per Square Inch.	F.°	Inches of Mercury.		Kilowatts.	Lbs. per Hour.	Lbs. per Kw.-hour.
126	0	28.0	5044	52.7	1480	28.0
132	0	28.5	4880	0	320	...

The above shows the results on small turbines.

For larger powers and with superheat the results are, of course, much better, as the following table shows:—

TABLE XIX.—VARIOUS 500-KILOWATT TURBO ALTERNATORS.

Pressure of Steam above Atmosphere at Stop Valve.	Superheat at Stop Valve.	Vacuum in the Turbine Cylinder. Bar. = 30".	Revolutions per Minute.	Load.	Steam used.	
			SCARBOROUGH ELECTRICAL SUPPLY COMPANY.			
Lbs. per Square Inch.	F.°	Inches of Mercury.		Kilowatts.	Lbs. per Hour.	Lbs. per Kw.-hour.
126	0	26.75	2400	529	22.7	12,023
128	0	27.7	,,	258	26.4	6,812
164	0	28.1	2600	0	...	1,477
			CHELTENHAM CORPORATION.			
130	0	26.7	3000	553	21.7	12,000
130	0	26.6	,,	278	26.2	7,280
133	0	24.0	,,	553	23.6	13,060
130	0	,,	,,	453	24.8	11,250
135	0	,,	,,	276	29.65	8,175
			BLACKPOOL CORPORATION.			
146	70	27.1	2500	515	21.35	11,000
150	0	27.0	,,	502	23.1	11,600
135	0	27.3	,,	497	24.0	11,953
133	66	27.3	,,	507	21.1	10,693
152	0	29.0	,,	0	...	1,500
160	0	23.6	,,	0	...	2,530
156	5	28.9	,,	0	...	1,465

It will be noted from the above results that the improvement in steam consumption resulting from a superheat of 50° F. is about 8 per cent., and from 100° F. it averages about 12 per cent.; also that for every inch of vacuum above 25 inches or 26 inches the consumption of steam falls about 4 per cent.

It will be noted that in steam turbines the steam consumption closely follows a right line law, or is proportional to the load *plus* a constant quantity which represents the consumption of steam at no load.

In connection with superheated steam it may be mentioned that, as there is no internal lubrication of the turbines, none of the usual difficulties which occur with reciprocating engines are met with in its employment.

Also, in steam turbines the absence of internal lubrication renders the exhaust steam absolutely free from oil, so that the water from the hot well can be returned to the boilers direct without oil filters.

Special tests have been made from time to time on turbine engines to verify the statement that no increase in steam consumption occurs with the age of the plant under fair wear and tear.

As to the consumption of steam on still larger plants, a long and exhaustive series of tests was made in January 1900 by Mr. W. H. Lindley and Professors Schröter and Weber, on behalf of the city of Elberfeld in Germany, on one of two 1000-kilowatt turbo alternators built at Heaton Works for that city. The turbo alternators were constructed to give 1500 kilowatts at 4000 volts 50 periodicity, the alternators being four pole running at 1500 revolutions per minute, and directly coupled to the turbines. The expansion of the steam was carried out in two cylinders, a high pressure and a low pressure, the steam being expanded down to a little below

the atmosphere in the first, and from that to the vacuum of the condenser in the second.

We have not space here to refer to the *modus operandi* of these tests, nor all the results. A summary of them will suffice.

TABLE XX.

Load.	Average observed Steam Pressure Absolute.	Corresponding Temperature of Saturated Steam.	Average observed Temperature of Superheated Steam at Inlet Valve.	Superheating (Col. 4–Col. 3).	Observed Steam Consumption per Kilowatt-hour.
(1)	(2)	(3)	(4)	(5)	(6)
Kilowatts.	Kgs. per cm^2.	C.°	C.°	C.°	Kgs.
1190.1	10.11	179.3	189.5	10·2	8.81
994.8	10.47	180.9	192.0	11.1	9.14
745.3	10.76	182.0	190.0	8.0	10.12
498.7	10.40	180.6	209.7	29.1	11.42
246.5	10.14	179.4	196.4	17.0	15.31
No load with excitation.	10.34	180.3	193.0	13.3	per hour. 1844
No load without excitation.	10.49	181.0	194.5	13.5	1183

For the following outputs in round numbers the steam consumption per hour is as shown:—

TABLE XXI.

Output.	Steam Consumption per Hour.	Steam Consumption per Kilowatt-hour.
Kilowatts.	Kilogrammes.	Kilogrammes.
1250	10,786	8.63
1000	9,189	9.19
750	7,496	9.99
500	5,707	11.41
250	3,821	15.28

On the second plant, tests were made to determine the advantages of superheating, and also the effect of varying the vacuum.

TABLE XXII.

Pressure Stop Valve.	Superheat.	Vacuum. Bar. = 30″.	Kilowatts.	Steam per Kilowatt-hour.
Lbs. per Square Inch.	C.°	Inches of Mercury.		Lbs.
157·5	0	26.97	1010	23.08
153	0	24.45	1041	25.25
125	0	27.10	1022	20.47

These show a gain of about 12 per cent. with 55° C. superheat, and that every inch of vacuum improves the consumption about 4 per cent.

In non-condensing plants also many tests have been made, but, as will be expected, the steam turbine compares rather more favourably with the reciprocating engine in condensing types. In a 100-kilowatt size a consumption of 39 lbs. per kilowatt-hour has been attained, and in a 250-kilowatt turbo dynamo 38 lbs. per kilowatt-hour, both with about 130 lbs. steam pressure and no superheat.

In larger sizes of 1500 kilowatts, with 200 lbs. steam pressure and 150° F. superheat, a consumption of 28½ lbs. per kilowatt-hour non-condensing has been guaranteed, and is expected to be easily attained, if not surpassed.

At the discussion of these results at the International Congress in Glasgow, 1901, Mr. Gerald Stoney added the following interesting remarks. He said: "Tests made on turbines which have been running for a long time showed that there was no falling off of economy. With reference to superheat, a small degree of superheat gave about 5 to 6 per cent. extra economy: 50° F. 8 per cent., and 100° F. 12 per cent. No tests had been made at higher degrees of superheat, but there was no doubt the economy would increase with higher superheats, and would probably reach 20 per cent. with 350° F. There was no difficulty in working with any degree of superheat the superheater would stand. The effect of a good vacuum was felt more in the steam turbine than in an ordinary engine, as it expanded right down to the vacuum of the condenser, which is not possible in an ordinary engine. In relation to the consumptions of ordinary engines and turbines at Elberfeld, where both were installed in the same station and therefore were tested under the same conditions, the consumption of the steam turbines from three-quarters to full load was better than that of the Sulzer triple-expansion engines."

These large plants are shown in Plate VII., where the high pressure and intermediate pressure turbines are at the left-hand side in one casing. The low pressure turbine is shown at the middle, with a bearing between it and the first two. Then comes another bearing, and beyond that the alternator and a small exciter.

The turbines of Mr. Parsons were tested on the ship *Turbinia* by Professors Ewing and Dunkerly for steam consumption. For first trial purposes the *Turbinia* was constructed—her dimensions being 100 feet in length, 9 feet beam, 3 feet draught of hull, and 44 tons displacement. She was fitted with turbine engines of 2000 actual horse-power, with an expansive ratio of 150-fold, also with a water-tube boiler of great power, of the express small tube type, but with no feed heater. The turbine engines consisted of three separate turbines—the high pressure, the intermediate, and the low pressure—each driving one screw shaft independently; to the low pressure or centre shaft the reversing turbine was also coupled, and on each shaft were keyed three propellers of small diameter and of normal pitch ratio. This arrangement was found to be the best after many trials, and has since been adhered to in subsequent vessels. The maximum indicated horse-power that has been obtained on runs of about 5 miles' duration has been 2300, giving a speed of 34½ knots; but a speed of 31 knots can be maintained for about 2 hours' duration, and as recently as last August runs at this speed were made at Havre for the Committee of the Paris Exhibition, the vessel at the time being heavily laden and with a foul bottom, showing that after four years of work the turbine engines do not deteriorate in efficiency. Since she was completed she has run several thousand miles, sometimes in very heavy seas, and the main engines have never caused a moment's anxiety, nor have any repairs to them been required.

The tests, which occupied more than a fortnight, were very elaborate, and comprised feed-water measurements, of great accuracy, at various speeds up to 31 knots. The water measurements were made by meters which were calibrated before and after each day's running. At the higher rates of speed two meters were placed in series so as to check each other; the errors as determined by the calibration were less than 2½ per cent., and the results were taken as accurate within less than this amount.

The horse-power was determined by model experiments in one of the principal testing tanks in this country.

At the speed of 31 knots the consumption of steam for all purposes was determined to be 14½ lbs. per indicated horse-power, the coefficient of propulsive horse-power to indicated horse-power being taken at 55 per cent. In other words, the consumption of coal for all purposes with a good marine boiler, under ordinary conditions and mild forced draught, would be less than 2 lbs. per indicated horse-power per hour.

The vessel's reversing turbine, which operated the centre screw shaft only, gave her an astern speed of 6½ knots, and when running at a speed of 30 knots she could be brought to rest in 36 seconds. It was also found that she could be brought from rest up to a speed of 30 knots in 40 seconds.

In these tests the indicated horse-power is given, but it is difficult to estimate this indicated power in turbines.

That such a small vessel could be fitted with power equal to 2000 actual horse-power strikingly shows the small bulk of such a marine turbine plant.

The De Laval tests were made by Messrs. Erick Andersson, Karl Wallin, and Axel Estelle, Stockholm.

The turbine dynamo with which the trial took place was provided with a centrifugal pump coupled to the turbine, and worked directly from one of the inductor shafts. This pump, at a suction height of 14 feet 9 inches, forced the water at a pressure of 11.5 lbs. per square inch through an ejector condenser.

The turbine dynamo was placed close to the boiler used during the trial. The steam pipe leading to the turbine was provided with a water strainer.

The boiler, which was tubular and provided with an inside furnace, was fed with water at a temperature of 43° Fahr., and generated steam at 118 lbs. per square inch, which was reduced, however, to 114 lbs. by the throttle valve of the governor.

The trial lasted eight hours without interruption, and the fuel used was "Best South Yorkshire Steam Coal," of which 1472 lbs. were consumed during the trials, while the steam consumption amounted to 9823 lbs., consequently the ratio of evaporation was $\frac{9823}{1472} = 6.67$. The vacuum was 13 lbs.

The temperature of the condensing water was raised by the steam from 34° to 50° Fahr., consequently the quantity of condensing water may be estimated at about 66 times the quantity of steam, or 1271 cubic feet per hour.

The dynamo developed during the whole trial an average electrical force of 113.4 volts × 324.0 ampères, or $\frac{113.4 \times 324.0}{736}$, equal to 49.92 electrical horse-power, while the number of revolutions of the inductors was 1506.

Hence the result of the trial gave a steam consumption of $\frac{9823}{8 \times 49.92} = 24.5$ lbs. per hour per electrical horse-power.

To ascertain the steam consumption at varying loads Andersson and Wallin made a second trial on the 4th March in the following manner:—

1. With the same steam pressure, 114 lbs., at the nozzle case—*i.e.* in the space between throttle valve of the governor and the steam nozzles—as on 15th February, and with all of the six steam nozzles in their places, the turbine dynamo was worked, and developed an electrical power of 113 volts × 326 ampères, or $\frac{113 \times 326}{736} = 50.05$ electrical horse-power, consequently the conditions were as nearly as possible the same as at the previous trial.

2. Then one of the steam nozzles was taken out and inserted into a separate steam

pipe leading from the nozzle case to a vessel containing a quantity of water, the weight of which had been ascertained. The vessel was placed on a sensitive balance, so that any increase in weight could immediately and accurately be ascertained. With the same steam pressure as in Sec. 1 the turbine dynamo was driven with five steam nozzles, developing an electrical effect of 113.5 volts × 264.2 ampères, equal to an average of 40.74 electrical horse-power. At the same time the stream, which during 12 minutes rushed through the nozzle mentioned above, was condensed and was found to be 40.5 lbs., or per hour $\frac{60}{12} \times 40.5 = 202$ lbs.

At the end of the trial all of the nozzles were carefully measured and found to be of exactly the same cross section, consequently the quantity of steam which passed through them was 6×202 lbs. $= 1212$ lbs. per hour. If the steam consumption in Sec. 1 is calculated at the same rate the result is $\frac{1212}{50.05} = 24.2$ lbs. per hour per electrical horse-power, which differs so very little from the result at the trial on 15th February during 8 hours, but the trifling difference may be owing to a slight variation in taking observations.

As the turbine dynamo, with five steam nozzles open, developed an electrical effect of 40.79 horse-power, and the calculated steam consumption was $5 \times 202 = 1010$ lbs. per hour, in this case the quantity of steam per hour per electrical horse-power would be $\frac{1010}{40.79} = 24.76$ lbs.

3. At this test 2 of the steam nozzles were closed, while the steam pressure remained the same, namely, 114 lbs. The turbine dynamo, with three steam nozzles opened, now developed an electrical power of 113.5 volts × 140.8 ampères, equal to 21.72 electrical horse-power, while the steam consumption, calculated at the same rate as in the previous case, was 3×202, equal to 606 lbs. per hour, or per electrical horse-power per hour $\frac{606}{21.72}$, equal to 27.9 lbs.

4. The turbine dynamo was then driven with four steam nozzles open, and the electrical load was regulated to 113.5 volts × 164.3 ampères, equal to 25.34 electrical horse-power, the steam pressure being reduced by the throttle valve of the governor to 93.8 lbs., with a vacuum of 13.27 lbs. The quantity of steam consumed was measured in the same manner as in Test II., and was found to be 174.2×4, equal to 696.8 lbs. per hour, consequently per hour per electrical horse-power $\frac{696.8}{25.34}$, equal to 27.49 lbs.

5. One of the steam nozzles was next closed, so that, as in Test III., the turbine dynamo was driven with three steam nozzles open, and the electrical load regulated down to 113.5 volts × 83.5 ampères, equal to 12.87 electrical horse-power, when the steam pressure was reduced to 74 lbs., with a vacuum of 13.5 inches. On measuring the quantity of steam consumed it was found to be $137.28 \times 3 = 411.84$ lbs. per hour; consequently per hour per electrical horse-power $\frac{411.84}{12.87}$, equal to 32.0 lbs.

In the tests with the 300 horse-power turbine the steam was superheated,—in the first case about 60° Fahr., and in the latter about 20° Fahr. It is evident that superheating is advantageous to the turbine, as it gives the steam jet a higher velocity and thus increases the kinetic energy of the steam, and it also diminishes the resistance of the turbine wheel, as illustrated by Table XXIV.

The use of superheated steam in connection with turbines has become more general in recent years. Practically any degree of superheating can be used, as the highly heated steam does not come into contact with the moving parts of the machinery; by

the time the steam reaches the chamber in which the turbine wheel revolves it has already the pressure and temperature of the exhaust steam.

TABLE XXIII.—RESULTS OF TESTS WITH DE LAVAL STEAM TURBINES AT DIFFERENT LOADS.

Turbine Machine.	Pressure of Admission Steam. Lbs. per Square Inch.	Vacuum Inches of Mercury.	No. of Nozzles open.	Electrical Horse-power.	Lbs. of Steam per Electrical Horse-power per Hour.	Remarks.
50 horse-power turbine dynamo. The test made in April 1895.	113.8	26.3	6	49.4	24.6	Work for condensing included.
	113.8	26.3	5	40.2	25.2	
	93.9	26.9	4	25.0	27.9	
	74.0	27.5	3	12.7	32.5	
100 horse-power turbine dynamo. The test made in June 1897.	103.7	25.8	5	92.7	22.6	Work for condensing not included.
	103.8	26.4	3	55.6	22.7	
	107.4	26.8	2	35.0	24.7	
	106.7	27.9	1	15.5	27.8	
				Brake Horse-power.	Lbs. of Steam per Brake Horse-power per Hour.	
150 horse-power turbine motor. The test made in November 1897.	113.8	26.4	7	163.0	17.6	Work for condensing not included.
	116.9	25.9	6	138.4	18.2	
	113.8	26.2	5	114.5	17.9	
	114.3	26.5	4	88.3	18.7	
	112.4	27.0	3	64.1	19.0	
	116.2	25.7	2	37.5	22.3	
300 horse-power turbine motor. The test made in December 1899.	192.7	27.3	7	303.6	14.1	Work for condensing not included.
	196.3	27.6	6	255.5	14.7	
	196.3	27.6	5	216.9	14.4	
	196.3	27.6	4	172.6	14.5	
	190.6	27.8	3	121.6	14.9	
	196.3	28.1	2	74.2	17.2	
	213.3	28.5	1	31.5	21.6	
300 horse-power turbine motor. The test made in June 1900.	126.6	26.98	8	337.45	15.68	Work for condensing not included.
	126.4	26.99	7	293.7	15.76	
	125.0	27.24	6	249.1	15.92	
	125.0	27.62	4	162.7	16.25	
	125.0	27.91	3	118.9	16.70	
	125.0	28.16	2	73.5	18.00	
	125.0	28.25	1	30.4	21.77	

From the results of trials it is obvious that not only the steam consumption but also the heat consumption (in thermal units per horse-power per hour) sinks with increasing superheating. With constant peripheral speed of the turbine wheel and increasing superheating the impact is increased on account of the higher velocity of the steam ; but this loss is instantly transformed into heat, which raises the temperature of the exhaust steam and thus diminishes the resistance of the turbine wheel.

It has been seen that in the construction of the De Laval steam turbine it is necessary to adopt a very high speed of the principal driving part of the machine. This speed is afterwards reduced in the machine by means of gearing, so that the turbine can be coupled to ordinary machinery.

TABLE XXIV.—TESTS WITH A 30 HORSE-POWER STEAM TURBINE WORKING WITH SATURATED AND SUPERHEATED STEAM RESPECTIVELY.

NON-CONDENSING—Steam pressure, 7 atmospheres absolute = 88.2 lbs. Speed of driving shaft, 2000 revolutions per minute. Speed of turbine wheel, 20,000 revolutions per minute.

		Half Load.		Full Load.	
		Saturated Steam.	Superheated Steam.	Saturated Steam.	Superheated Steam.
Temperature of the Steam	Centigrade	164	460	164	500
	Fahrenheit	327	860	327	932
Power developed	Metrical brake horse-power	21.4	24.5	44.1	51.9
	English brake horse-power	21.1	24.2	43.5	51.2
Steam consumption per brake horse-power per hour	Kilogrammes per metrical brake horse-power	21.6	14.1	17.7	11.5
	Lbs. in English brake horse-power	48.3	31.5	39.6	25.7
Heat consumption per metrical brake horse-power per hour in metrical heat units		14,160	11,270	11,610	9390
Temperature of exhausted steam	Centigrade	100	309	100	343
	Fahrenheit	212	588	212	649

The combination of the De Laval turbine to other classes of machinery is receiving consideration, yet experiments in this line are not sufficiently advanced that these machines can be put on the market.

The De Laval type of turbines excel in smaller units, while the Parsons' types are essentially large power machines, and it is significant that NEW turbines are made of large units before any public tests are given.

Small steam engines of any kind are disappearing before the gas and oil engines, so that the prospect before steam engines is for large units in which they can at present compete against the internal combustion engine.

There is no doubt the steam turbine has given a new lease of life to steam as a power working fluid, and that they will altogether supersede the reciprocating engine for all purposes where steam is used is beyond any doubt now.

And all indications point to the culmination of all the improvements in heat engines in the evolution of an internal combustion turbine.

Attempts have been made towards this desirable end. Mr. Parsons in his original patent proposes an internal combustion turbine.

Mr. Ferranti, in Specification No. 2565, 1895, proposes the same thing. A brief description from this specification will show the nature of the proposal :—

In one development of the invention air is used and compressed by means of pumps through a chamber lined with refractory material and capable of standing considerable pressure which contains the combustible, preferably in the form of coal ; the working fluid then becomes heated air, and the products of combustion leaving the fire chamber at a very high temperature and expanding through expansion tubes so acquire

the necessary velocity and diminish in temperature accordingly. The working fluid is then used in an impact reaction engine.

A portion of the exhaust gases may be used which are compressed with fresh air through the fire chamber. This is desirable in some cases, so as not to get such extremely high working temperature, the proportion of air and burnt gases being settled by experiment according to the temperature at which it is desired to work as a maximum.

In some cases which are worked with a nearly closed cycle the gases exhaust from the reaction engine at a pressure of several atmospheres. They are then, by preference, passed through a re-generator and cooler, after which they are pumped through the re-generator and the fire chamber together with the necessary amount of fresh air to support combustion.

In another form the fire chamber burning solid fuel is replaced by a chamber into which powdered coal or oil is injected together with the air for the purpose of combustion, and to act as a working fluid; the chamber is kept under a high pressure, combustion taking place immediately and completely, the products issuing at a high velocity

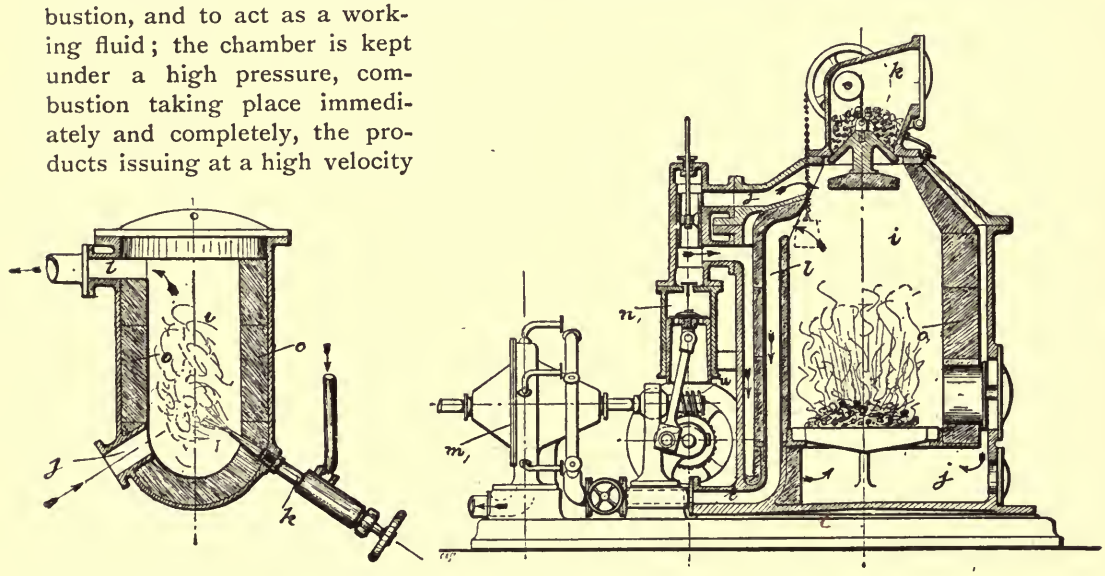


FIG. 227.—Ferranti's Hot Gas Turbine.

through expansion tubes and then giving up their energy to the impact engine. In this case a portion of the products of combustion may also be introduced with or without cooling, and with or without a regenerator into the fire chamber. Coal or other gas may be used for compression, and injected in place of the other combustible into the fire chamber together with the same or other modifications in the cycle above described.

In some cases the high speed reaction engines are combined with the necessary pump gear driven for compressing the working fluid through the fire chambers. The gearing is so arranged that the high speed of rotation of the motor is geared down to a sufficiently slow speed to work the reciprocating compressors.

When air and the products of combustion are used as a working fluid, the speed and power are regulated by the amount of the fluid which is compressed through the fire chamber, the pressure being kept constant, or as nearly so as possible, by bringing more or less expansion tubes into play to act upon the wheel, the object of this being to always work with the maximum range of temperature and so secure the maximum economy at low loads.

When working with steam condensing impact engines or turbines, ejector con-

densers are used, in which the necessary power required outside the engine contained in the steam is supplied by means of a high pressure jet of water generated by a high speed centrifugal pump run direct by the axis of the reaction engine which supplies a small jet of water which acts by induction on a larger jet of water in the condenser.

In some cases a water spray or cooling jackets may be used in the air compressors, but where such is the case it is desirable so far as possible to separate the moisture so that it should not pass into the combustion chamber.

Fig. 227 shows a combination of hot air furnace, compressing pumps, and reaction engine; and a fire chamber arranged for burning gas, oil, or other suitable fuel is shown in the left-hand Figure.

At Fig. 227 (*i*) is a hot air chamber, (*j*) is an entrance for compressed air, (*k*) is an arrangement for introducing the fuel, (*l*) is the exhaust for the burnt and heated gases to the reaction engine (*m*), (*n*) is an air compressor.

The air pump is shown driven by a worm and wheel gear.

None of these proposals are practicable. It will be observed that the working pressure can never exceed the pumped air pressure; it is the volume of the gases which are increased by the heat. The same proposals were made half a century ago for reciprocating piston engines by Joule. The fluid is taken in at one volume and rejected at a larger volume, but the expansion takes place mostly in the furnace, and the power is proportional to the difference in area between the pump piston and the engine piston.

The most successful engine of this type working on the reciprocating principle is that known as the Bucket Engine, described in Vol. II. of this work. I have worked upon the idea of the internal combustion turbine or hot gas turbine myself for some years, and my own conclusion on the matter is that the combustion must not be continuous; and the engine must be worked by a fluid which is taken into the combustion chamber at one pressure, and sent out at a higher pressure to expand in the turbine. Experiments led to the adoption of a pressure generator, as shown in Fig. 228, which is a diagram drawn to explain the invention, without regard to details of construction.

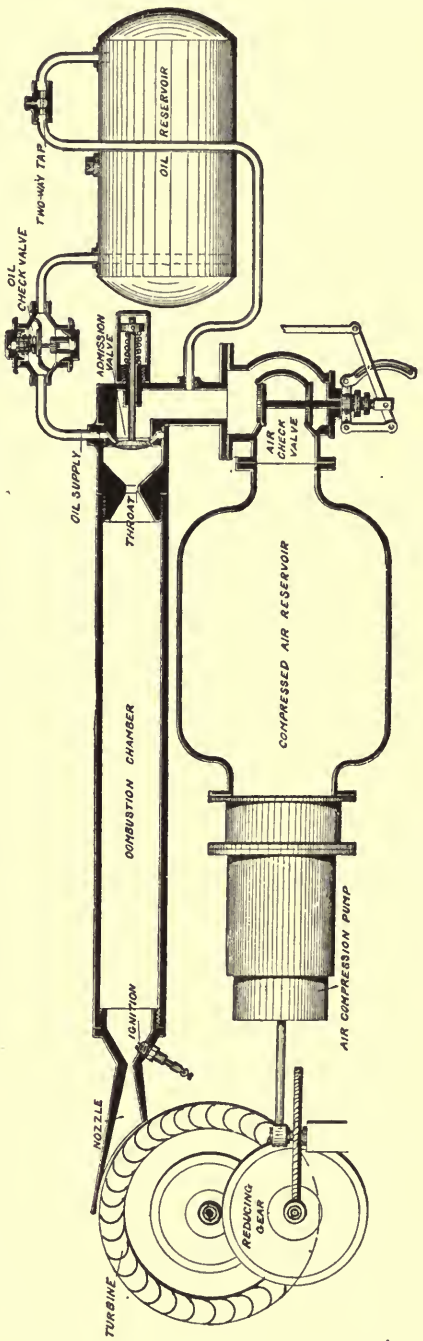


FIG. 228.—Gas Combustion Turbine.

It consists of a turbine with nozzles as usual,—a combustion chamber into which the air and fuel are forced or drawn, an ignition plug to fire the mixture. In the diagram the air and oil fuel enter the combustion chamber by one valve, and pass through a throat,

in order that the fresh incoming charge should clear out the remains of the previously fired charge.

The air and oil may be drawn in by a suction fan on the turbine exhaust much as they are drawn into a motor car petrol engine cylinder, but it is preferable to force them in under pressure. A compression air pump and reservoir is therefore provided, and the air used at from 50 to 80 lbs. pressure. The oil reservoir is connected to this same air reservoir, so that the oil and air are at equal pressures at the admission valve.

The action of the apparatus is as follows :—

The reservoir of compressed air is opened by the check valve. Air and oil enter by the admission valve to the combustion chamber. The ignition plug is sparking; immediately the mixture reaches the ignition the mixture fires, and the increase of pressure closes the admission valve. The hot gases under pressure rush through the nozzle and drive the turbine. As soon as the pressure falls below that of the air reservoir, air and oil again automatically enter the admission valve, driving the burnt gases before them until they reach the sparking plug, when another combustion occurs. The valve again closes, and the turbine obtains another impulse, and so on the cycle goes intermittently.

The air pump is driven by gearing from the turbine, and the half of the combustion chamber near the throat is cooled by water jacket. The nozzle is also water jacketed.

In some cases the sparking plug has been tried about the middle of the chamber, in order to present an elastic gas cushion between the fired gases and the nozzle, and to prevent flame passing through the nozzle.

The necessity for water cooling, of course, reduces the efficiency much in the same degree as in the reciprocating engine, although in the turbine a higher temperature of the working fluid is permissible.

The continuous combustion generator is best adapted for solid fuel, coal, or coke, while the intermittent system is best for gaseous or liquid fuel.

THE DESIGNING OF TURBINES

No detail designs for steam turbines have as yet been published. Patent specifications only give the principles and methods, and generally the designs are as far from actual practice as the Patent Law will allow. Only by actual dissection of working turbines can the actual construction be found. For the benefit of engineers who may desire to have a more intimate acquaintance with turbines' design and construction, I shall here conclude this chapter by a design which has been made and tried by myself. There are many who may find a use for small steam turbines, easily and cheaply made, not of transcendental economy in steam, but as good as small steam engines reciprocating, and of same output of power.

Two types of turbines were chosen : first, the Pilbrow double wheel type, described in his Specification No. 9568 of year 1843, and shown in diagram Fig. 163, the two wheels running in opposite direction and coupled by gearing.

The Seger steam turbine, illustrated in Fig. 210 (p. 169), is of this type, the gearing being belting.

The proper form of the passages in the two wheels is shown in the diagram Fig. 162. The steam enters from nozzles converging on the first wheel, into which it flows at a pressure about 10 to 15 per cent. less than that due to the boiler, with a velocity of about 1500 feet per second. It thus drives the first wheel by impact and pressure. It expands in passing the first wheel, and acquires more velocity up to about 2000 feet per second. It then enters the second wheel, and by impact drives that wheel in the opposite direction. Many attempts have been made to carry this action further, with a view to reducing the speed of the wheels, by adding more wheels in series, but such a practice is fallacious.

With these partial flow wheels two wheels only give the maximum results. All other arrangements are contrary to theory, and obviously bad. If we want slower velocities to deal with, then the second type of turbine, herein described, must be adopted, that described by Wilson in his Specification, No. 12,060 of year 1848, and perfected by the Hon. C. A. Parsons, and now well known.

There is no getting away from the fact that either one or the other type is the only choice for simple construction and good performance.

If we attempt to put up, say, a series of De Laval wheels we meet with immense mechanical difficulties, not to speak of steam distribution difficulties. In large size turbines these difficulties, by high-class tools and workmanship and fine designing, can be and have been overcome to a large extent in machines over 100 horse-power. Engineers who make turbines over 100 horse-power can generally get all the designs

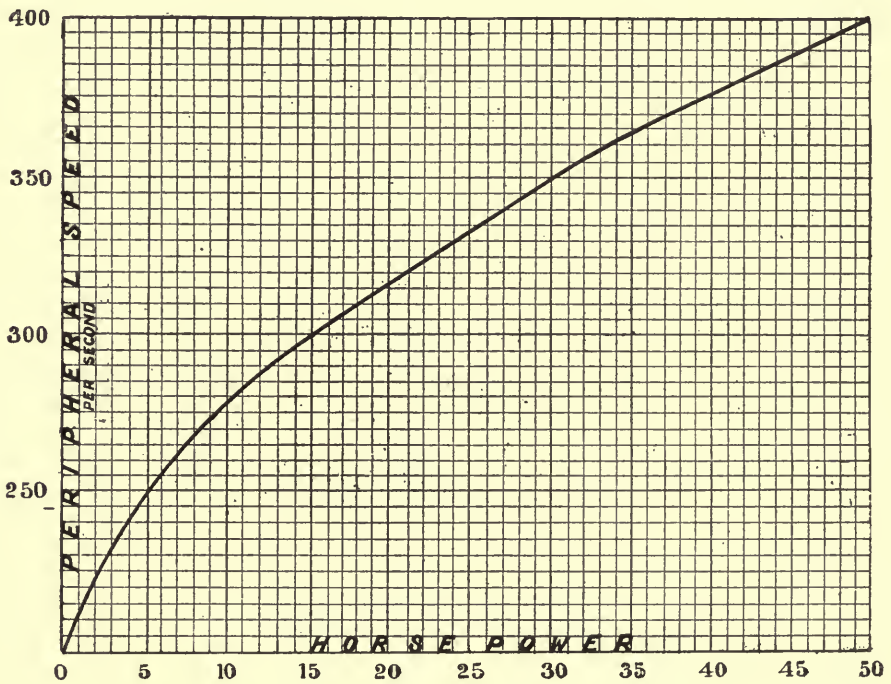


FIG. 229.—Turbine Speed Curve.

they require from a highly trained and paid staff. The designs now under consideration are intended as instructive and useful to those who may have to make their own designs and work out smaller affairs.

Taking the first design, the two wheels are our first concern. We shall assume a turbine is required for 5 horse-power, steam pressure 100 lbs. at the boiler, non-condensing. It will give much more power condensing, and may so be used by arranging a proper packing for the shafts, to prevent air being drawn in.

In this class of turbine, with a single wheel, a peripheral speed on the middle diameter of the buckets is less the smaller the turbine is, owing to the weakness of smaller wheels in comparison with larger ones. Thus a De Laval 5 horse-power wheel is made for 515 feet peripheral speed per second, while the 300 horse-power machine runs at 1378 feet.

The revolutions in the first case are 30,000, and in the latter 10,600 per minute.

As the turbine under discussion has two wheels, each to run at half the velocity of a single wheel, I have therefore, from the published results of the De Laval wheels and their

speeds, which have been found in practice satisfactory, made a curve for double wheels (Fig. 229) which shows the relation between horse-power and peripheral speeds at half the revolutions of a single wheel, the 5 horse-power machine having a speed for each wheel equal to 250 per second.

TABLE XXV.—TABLE SHOWING VELOCITIES AND HORSE-POWER OF TURBINE WHEELS—NON-CONDENSING. (*Theoretical.*)

Initial Steam Pressure, Lbs. per Square Inch.	Counter-Pressure, 1 Atmosphere.			Initial Steam Pressure, Lbs. per Square Inch.	Counter-Pressure, 1 Atmosphere.		
	Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy, Foot-lbs. per Second.	Horse-power of 550 Foot-lbs. per Second.		Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy, Foot-lbs. per Second.	Horse-power of 550 Foot-lbs. per Second.
		Per Lb. of Steam per Hour.				Per Lb. of Steam per Hour.	
60	2421	25.29	0.046	160	2992	38.63	0.070
80	2595	29.06	0.053	180	3058	40.35	0.073
100	2717	31.86	0.058	200	3115	41.87	0.076
120	2822	34.37	0.062	220	3166	43.26	0.079
140	2913	36.62	0.066	280	3294	46.83	0.085

There being two wheels running in opposite directions at half-speed, 250 each, we may calculate the steam required at the same rate as for one wheel going at 500 revolutions per second, the object of the two wheels being to reduce speed only.

From Table XXV. we find that with 100 lbs. pressure and the theoretical velocity 2717 per second the horse-power is 0.058 per lb. of steam, so that if the turbine is to be 5 horse-power the consumpt of steam would be $\frac{5}{0.058} = 86$ lbs. per hour.

But we cannot run wheels at such high speeds in practice. The strength of material limits the speeds, and the 5 horse-power wheel comes out at 4 inches diameter at the mid-line of the buckets. The peripheral speeds are given in the curve (Fig. 229) up to 50 horse-power, and these speeds multiplied by a constant, which appears to be 2, gives the maximum revolutions of a single wheel. This constant will then be equal to 1 for double wheels at maximum speed.

The steam required is inversely proportional to the peripheral velocity; hence we have found that, as 86 lbs. per hour are required at the high theoretical speed, we can find what is necessary at the lower practical speed, 500. We will reduce the steam consumption to lbs. per second, $\frac{86}{3600} = .024$; then $500 : 2717 :: .024 = 0.1304$ lbs. per second, 472 lbs. per hour.

We have seen that the flow of steam at its maximum is $Q = 370\ AD$ in lbs. per minute, where A is the area of the nozzle at the narrow end; hence $\frac{Q}{370 \times D} = A$.

Q in this case equals 8 nearly, therefore $\frac{8}{370\ D} = A$, and D the density of steam at 100 lbs. pressure $= 0.23$. Therefore $\frac{8}{370 \times 0.23} = 0.09$ square inch, equal to one jet of $\frac{11}{32}$ inch diameter, or two jets each of about $\frac{5}{32}$ inch.

The quantity of steam seems large, but for a small non-condensing engine at 100 lbs. boiler pressure it is by no means uncommon. The turbine gains more by condensing than a reciprocating engine, so that probably half the steam would be saved by condensing.

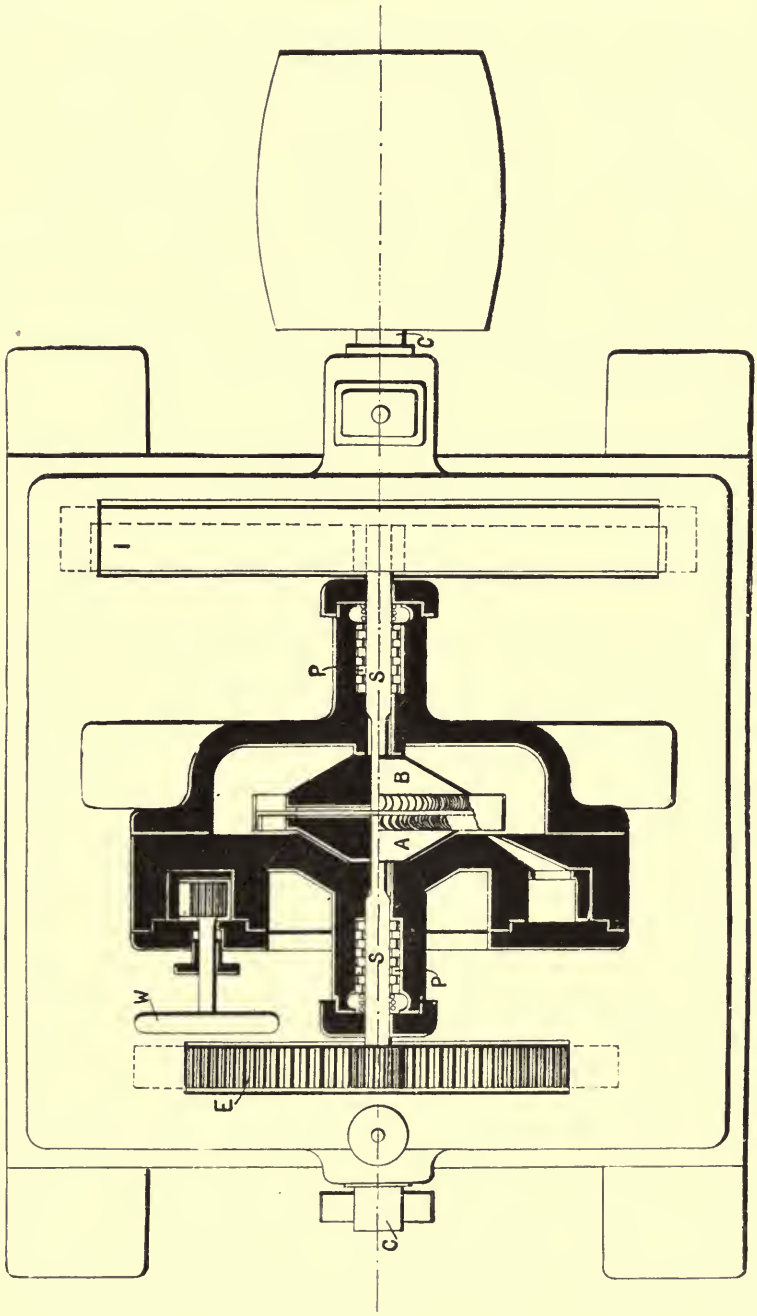


FIG. 230.—Sectional Plan of Double Wheel Turbine. Scale, ¼ actual size

The jets would be reduced in area by one-half for condensing, as may be gathered from the following table, in which it will be seen that the horse-power is doubled at 100 lbs. pressure, exhausting into a 28-inch vacuum :—

TABLE XXVI.—TABLE OF VELOCITIES HORSE-POWER EXHAUSTING INTO VACUUM.

Steam Pressure, Lbs. per Square Inch.	Counter-Pressure 0.93 Lbs. per Square Inch absolute, corresponding to 28-inch Vacuum.			Steam Pressure, Lbs. per Square Inch.	Counter-Pressure 0.93 Lbs. per Square Inch absolute, corresponding to 28-inch Vacuum.		
	Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy, Foot-lbs. per Second.	Horse-power of 550 Foot-lbs. per Second.		Velocity of Outflow of Steam, Feet per Second.	Kinetic Energy, Foot-lbs. per Second.	Horse-power of 550 Foot-lbs. per Second.
		Per Lb. of Steam per Hour.				Per Lb. of Steam per Hour.	
60	3680	58.44	0.106	160	4045	70.61	0.128
80	3793	62.08	0.113	180	4091	72.22	0.131
100	3871	64.66	0.118	200	4127	73.50	0.134
120	3940	66.99	0.122	220	4159	74.64	0.136
140	3999	69.01	0.125	280	4229	77.18	0.140

We have now seen that the wheels' diameter must be calculated at the velocities as given by a curve (Fig. 229), and the area of the nozzles is calculated from the steam required per second; and having found these, the design becomes a mechanical problem.

This double wheel turbine, as designed, is shown, $\frac{1}{4}$ actual size, in the three Figs. 230, 231, and 232.

Fig. 230 is a sectional plan, showing the two wheels A and B carried on flexible-ended shafts SS.

Fig. 231 is a side elevation, showing the base, the wheel gearing, and governor.

Fig. 232 is an end view, showing gearing wheels, governor, and regulating wheel for cutting off nozzles, and governor driving band.

Referring to Fig. 230, the special features are the gearing, consisting of an external and internal cut toothed wheel E and I. By this device the two turbine wheels are geared to one counter-shaft, and their speed reduced and power added.

If the large wheels are made of raw hide leather, compressed into gun-metal shrouds, the wheels are both silent and durable. The wheels are made heavy in order to act as buffer fly-wheels, so that in the event of a sudden heavy load being applied, such as a short circuit on a dynamo driven by the machine, the inertia of the wheels would absorb the shock before it could be transmitted to the thin flexible shafts SS.

The metallic packing at the bearings PP is specially designed to work tight, either on pressure or vacuum. Rings on the shaft made of anti-friction metal slip thereon. A good fit between these rings are cast iron or steel spring rings, which fit the bore closely; a spiral spring keeps the rings in contact sideways. These packings are lubricated by gravity feed. In the practical design of larger engines, bearings are placed beyond these packed bearings to take the strains up.

The nozzles are bored in the body of the casing, and regulated by a cover ring sliding over them, and which is moved by hand wheel W and a rack and pinion, shown in Fig. 233.

In Fig. 231 the counter-shaft C is seen passing through the base, as also is the governor G fixed to the stop valve V and driven by belt.

This governor can be also driven by direct connection to the shaft C, and the valve

operated by a simple lever. The exhaust is connected to the end cover, preferably at the bottom.

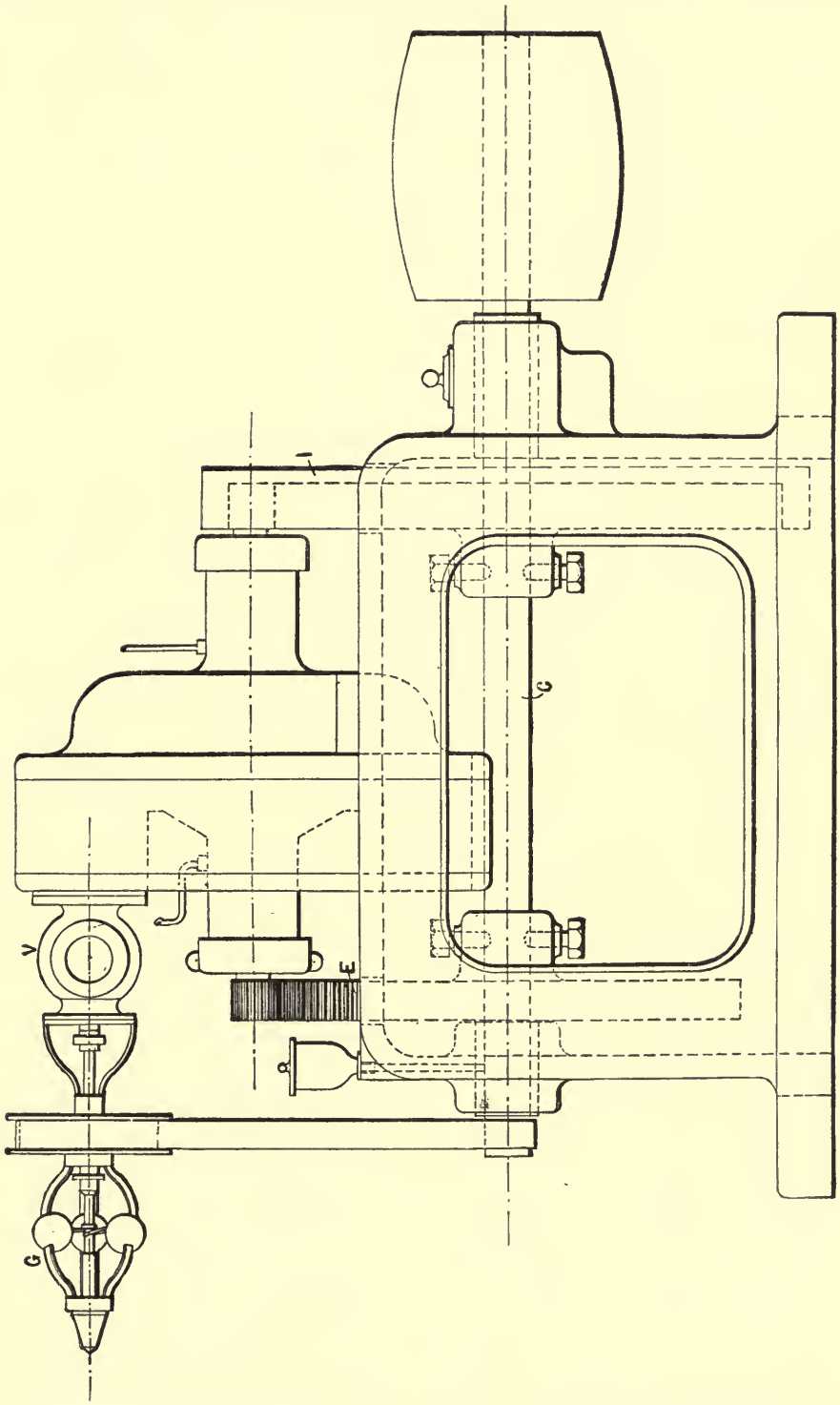


FIG. 231.—Elevation of Double Wheel Turbine. Scale, ¼ actual size.

The turbines work extremely well, and without attendance of any special kind.

Recently an improved system of forming buckets on the wheels has simplified the construction (see page 168).

The sliding cover ring for regulating the nozzles is shown in Fig. 234, where it will be seen that with three nozzles the cover has one hole same size as nozzle inlet, another hole twice the length, and another thrice the length of the first, so that by sliding the ring the holes may be closed in succession. This may be done by a governor and relay in larger turbines.

This type of turbine is intended for powers from $\frac{1}{2}$ horse-power to 50 horse-power.

For larger powers the second design comes in. As it has been fully described in the foregoing pages, under the description of Parsons' turbines, we need not go into the details.

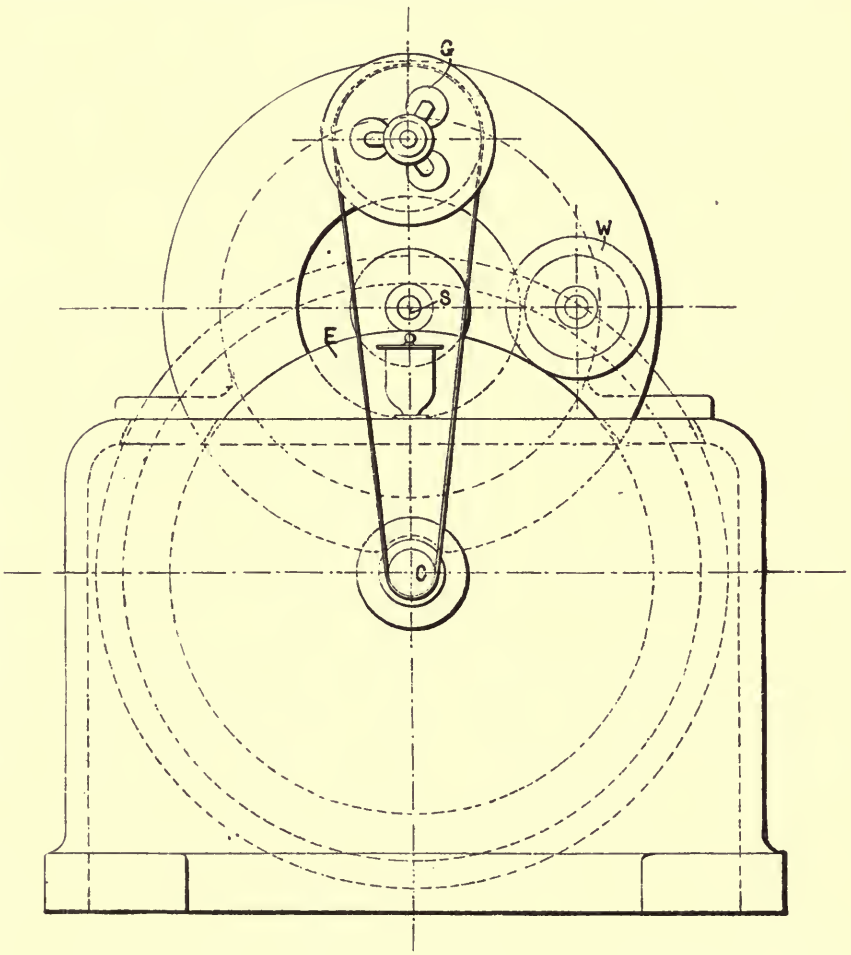


FIG. 232.—End View of Double Wheel Turbine. Scale, $\frac{1}{4}$ actual size.

Theoretically, the difference in pressure between wheel and wheel in a series of wheels should be very small. P is the pressure at the inlet side, and P_1 the pressure at the outlet side.

P_1 should be considerably greater than $\frac{3}{5}$ of P. The maximum flow of steam is obtained when $\frac{P_1}{P} = 0.577$, and $P_1 = 0.577 \times P$. Consequently if we had a nozzle like that shown in Fig. 108, with steam at 100 lbs. pressure entering at the wide end, the pressure at B, just beyond the throat, would be $P_1 = 0.577 \times 100 = 57.7$ lbs.

The pressure in one wheel should, in fact, differ by only a few pounds from that of the next; the quantity of the steam passed will then be $Q = 1.75\ A\sqrt{(p - p_1)p_1}$, wherein

Q = weight of steam per minute.
A = area of passages in square inches.
p = absolute pressure inside one wheel.
p_1 = ,, ,, ,, the next wheel.

Thus if $A = 0.1$ square inch, and $p = 100$, and $p_1 = 95$ lbs. per square inch, then $Q = 1.75 \times 0.1 \sqrt{(100 - 95)95} = 3.7$ lbs. per minute, or a quantity Q_2 of 16 cubic feet of steam. From this the velocity can be found in feet per second $= \frac{Q_2 \times 144}{60 \times A} = \frac{144 \times 16}{60 \times 0.1}$ $= 384$ feet per second, from which it will be at once seen that the velocity of the wheel can be brought much nearer the theoretical at not a very great velocity.

But this small drop in pressure can only be obtained by a very long series of wheels, with very small blades and close together: conditions easily enough met in large turbines, but very difficult in small ones. Hence in this class of turbine it is not advisable to make them below 20 or 25 horse-power.

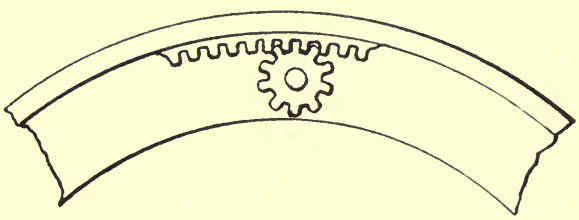


FIG. 233.—Rack and Pinion of Double Wheel Turbine.

The chief cause of their inefficiency in small sizes is due to the clearance spaces required for free rotation and the consequent leakage of steam past the blades. The clearance is much greater in proportion to the blade area in a small turbine than it is in a large one.

The problem is to design a wheel with blades easily made and with very many blades of thinnest material on the wheel, the blades to be increased in area by increase of radial length, and to be very narrow axially.

In order to make wheels to meet the requirements, a plain disc is turned up true and with a rim (Figs. 208, 209). The disc is slotted out all around, forming slots into which the blades are firmly driven; these blades are, at first, plain steel stampings; the upper part is then bent into shape. They are then inserted in radial slots in the wheels, and the lower part flattened back tangentially, thus firmly fixing the blades; they are afterwards further fastened by rivets. In this way a light strong wheel is formed. It is important for economy of steam that the wheels should be narrow and the blades numerous and thin. This design meets these requirements.

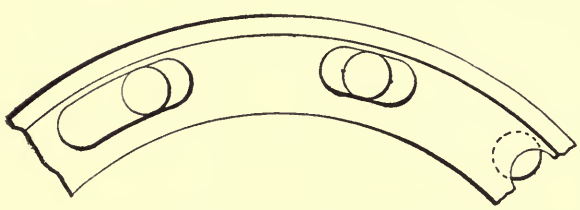


FIG. 234.—Nozzle Regulator of Double Wheel Turbine.

CHAPTER IV

ROTARY PISTON ENGINES

This class of engine is but little used as a prime mover. It is, however, used to some extent as a reversed motor for pumping water and blowing air. And it would seem that, even in these humble capacities, it is to be superseded by the free-running centrifugal or screw pumps for both air and water. However, no work on prime movers and

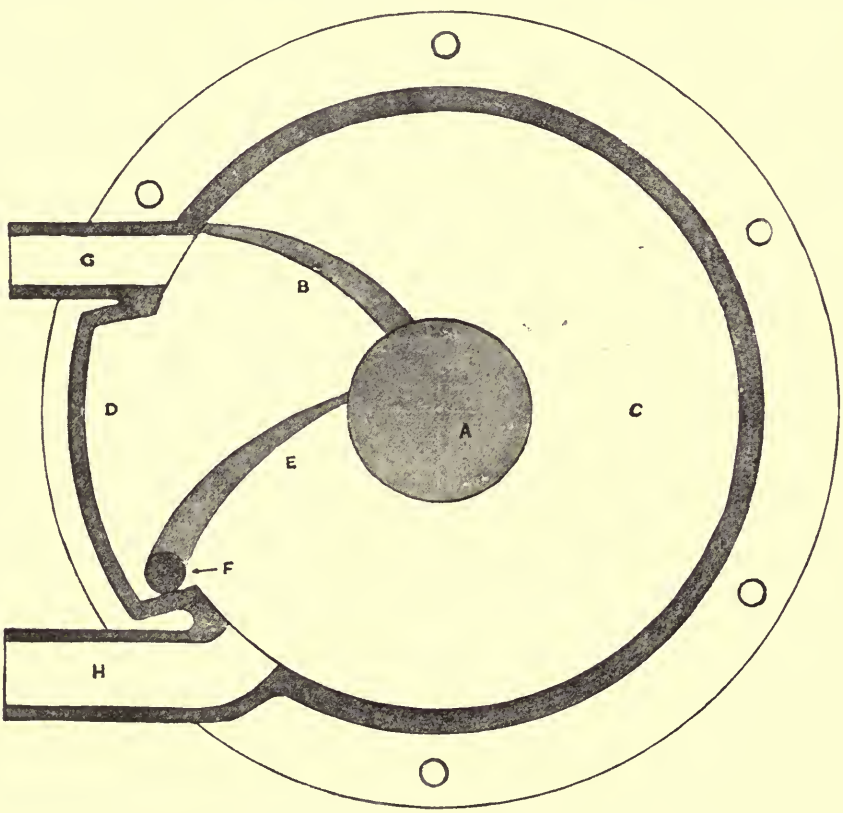


Fig. 235.—Diagram of Rotary Piston Engine.

modern engines would be complete without including an example or two of the best of the class and latest developments in their construction.

Perhaps more brains and money have been spent on the invention of such engines than on steam turbines, and yet they have never reached any practicable design

approaching that of the turbine. Most middle-aged engineers can recall to mind many brave attempts to run rotary engines successfully, all introduced with high hopes and determination to succeed, all with same discouraging end. I can remember in my youth two vessels fitted out with rotary engines in Scotland by a determined inventor, who fully believed he had solved the difficulties. They both sailed away, never again to be heard of: an impressive fact, which forcibly sent home the lesson that no uncertain experiments should be made on sea-going vessels.

The mistaken idea underlying nearly all the inventions in this direction seems to be that, by substituting a rotary piston for a reciprocating piston, the to-and-fro motions will be abolished, and therefore only smooth regular rotary motion obtained, with all the advantages of such a regular turning movement. But if one turns to the patent records of these inventions and examines them all carefully, in every case wherever an engine has been designed at all capable of passable working for a time it will be found that there are quite as many reciprocating parts, to-and-fro movements, as in a simple piston and crank engine.

It will also be found that the pistons, or doors, or sliders in the rotary are subjected to great rubbing friction at high speed, causing unequal wear and tear and eventual failure.

One diagram (Fig. 235) will illustrate the whole class. A is a shaft running through a cylinder C and supported in stuffing boxes in the cylinder covers, between which covers it fits steam-tight without end-play. It has a wing B, which forms the rotary piston. This wing must fit steam-tight at the ends and at the periphery, and many ingenious devices have been devised to pack this piston tight. E is a door or shutter, pivoted at F, so that it can shut up into D, to clear the piston when it comes round. G is the steam inlet and H the exhaust pipe. Theoretically this looks all right, and, beginning with this simple conception, the mechanical inventor adds improvements and alterations innumerable, and produces an engine which has all the defects of the simple affair sketched, only toned down and made partially tolerable.

BROWN'S ROTARY ENGINE

In Fig. 235 the wing piston is shown fixed to the shaft, but it may be hinged to it, so that the steam will press it tight at the periphery. Some inventors make the piston oval or eccentric. We shall, however, only consider briefly the last and best attempt on a practical scale, as described by Professor Jamieson. Fig. 236 is a cross-section and sectional plan of this engine made with double doors. The following is an index to the parts:—

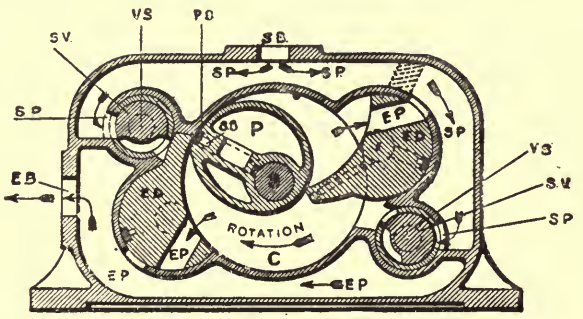
SB =	Steam branch.	PS =	Packing strips.
SP =	,, ports and passages.	SS =	Spiral springs.
SV =	,, valves.	S =	Shaft.
VS =	Valve spindles.	ED =	Exhaust door.
C =	Cylinder.	EP =	,, ports.
P =	Piston.	EB =	,, branch.

The steam enters by steam branch SB into the steam passage SP.

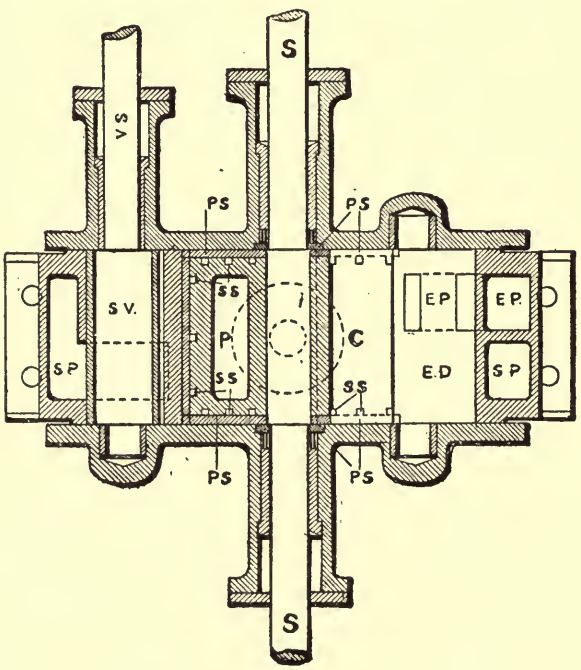
This steam passage thus forms a steam jacket to half the cylinder. It may be arranged so as to form also a "water trap" (with a return "loop" to the boiler), so that nothing but dry steam shall enter the cylinder. The underneath side of the cylinder is steam jacketed by the exhaust steam. The outside of both cylinder covers and jackets are carefully lagged with felt, and covered with wood to prevent radiation.

The steam then passes through the steam port in the *left*-hand revolving steam valve SV, into the cylinder, whenever the piston P has passed the nose of the *left*-hand oscillating exhaust door ED. The piston is thereby forced round under full steam pressure until it reaches the point of "cut off," when the *left*-hand rotating valve closes the steam port. For the remainder of the stroke the piston is propelled by the natural

expansive force of the steam (enclosed between it and this steam valve) until release takes place through the exhaust port EP, in the boss of the *right*-hand exhaust door ED; from whence the steam passes along the exhaust passage EP, and through the exhaust branch EB, to the atmosphere or to a condenser, according as the engine is working as a noncondensing or condensing one. Precisely the same action takes place during the remaining half of the piston's revolution by aid of the *right*-hand steam valve and the *left*-hand exhaust door.



Cross Section.



Sectional Plan.

FIG. 236.—Brown's Rotary Engine.

Nothing could be simpler under the circumstances, or more effective; for full advantage is taken of the expansive properties of steam by so arranging the "lead" of the eccentric (which turns the steam valves) that the "cut off" may take place at any desired proportion of the piston's stroke; or the "cut off" may be directly varied according to the load and steam pressure by connecting a loose eccentric to the governor.

It will be observed that the front edges and surfaces of the exhaust doors make a steam-tight fit with the circumferential surface of the piston, and that when these doors are closed they form part of the bore of the cylinder. They are forced

forward by the incoming steam pressing behind them, whenever the peculiarly shaped piston has passed them, and they are pushed *gently* back into their seats by the piston (during expansion and release) against the reaction of the pent-up or "cushioning" steam retained between them and the steam valves.

The working surfaces of the piston, steam valves, and exhaust doors are all packed by gun-metal strips, kept up to their bearings by small adjusted spiral springs. From the perspective view of the piston (Fig. 237) it will be seen that the side and end packing strips are checked into each other, as well as the side strips into the ring ones next the shaft. In order to make the wear uniform on the side packing strips the spiral springs are made proportionately shorter the farther they are from the centre of the shaft; and, since each of these springs are contained in the same length of hole, their outward pressure on their strip is inversely proportional to their radial distance from the centre of the shaft; or, in other words, inversely to their travel.

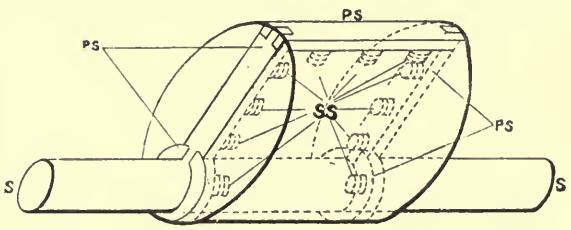


FIG. 237.—Brown's Piston.

BROWN'S SINGLE-DOOR ENGINE

From the description which we have just given of the double-door engine the construction and action of the single-door one will be easily understood from Fig. 238. This form of engine has not been steam jacketed, as it is intended for rougher and smaller work than the double-door one, such as driving fans, etc.

The points claimed for the engine are :—

1. Packing strips that are expected to wear uniformly and last a long time without requiring adjustment or renewal.

2. Early cut off with corresponding economy, due to taking full advantage of the natural expansive property of steam.

3. A minimum of clearance space.

4. Jacketing.

5. A thoroughly well balanced strong engine, occupying a minimum of space for the power developed, freedom from undue vibration and noise, combined with high speed and an economy of steam that has not been reached by any other rotary engine, or even by any simple reciprocating engine of the non-condensing and non-compound type.

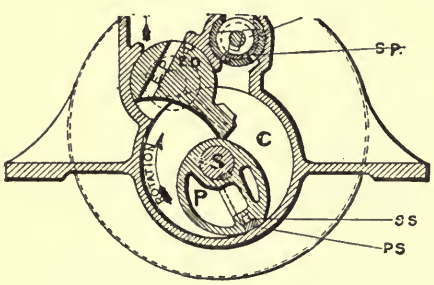


FIG. 238.—Single-Door Engine.

6. The possibility of compounding his engine, or even of adopting triple or further multiple expansion, and of using the highest practicable pressures.

The high speed of the rubbing points of the doors, and that under great pressure, is obvious, even in this otherwise excellent design, but the number of reciprocating parts is great, and it did not come into practical use.

THE ROTA ENGINE

In order to get over the difficulty of this high speed friction between the piston and cylinder or piston and doors a promising idea was patented by Mr. Macewan Ross, in

which the cylinder is made to revolve with the pistons, so that their relative speeds are much reduced. He called the engine the "Rota" engine. Figs. 239 and 244 represent the engine in sectional elevation. It consists of a cylindrical casing A with closed ends or covers, through whose axis from either end pass two central hollow drums B fixed to and carried by their respective brackets XX, and having cast in them ports for admission and exhaust of steam at both ends. The drums B (Fig. 244) are

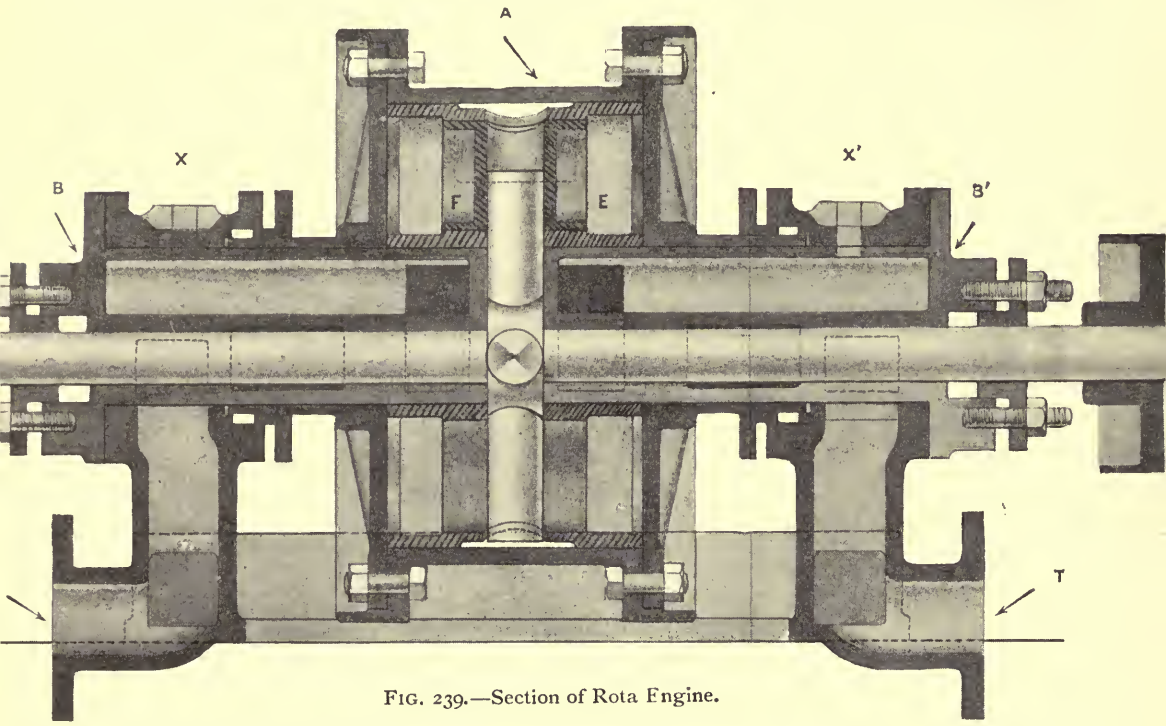


FIG. 239.—Section of Rota Engine.

divided by central webs running their entire length to form admission and exhaust passages *bb'*, from which the inlet and exhaust ports CC' are carried to the interior of the casing. A series of segmental pistons EE are fitted between the casing A and drums B, which fill up to a greater or less extent what forms the steam space within the casing.

The crank shaft D (Figs. 239 and 240) passes through bearings bored eccentrically in the drums, and is made with four arms or cranks which also act as slides, and which are fitted into slide blocks F fitted into their respective pistons. The pistons are thereby held in equal and equidistant angular positions relatively with the axis of the main casing A; the slide blocks in the pistons are allowed to oscillate in order to accommodate themselves to the different angular positions of the cranks, which reciprocate radially through them relatively with the axis of the main casing, as the distance of the piston from the axis of the crank shaft varies according to their position round the drum B. Their distance apart also varies, those pistons bearing on the portion of the drums having least eccentricity being close together, and those on the opposite side being farthest apart. When the pistons travel round the drum their speed is highest at the point of greatest eccentricity, and the ports CC' are formed to admit steam between each pair of pistons as they pass the point of least eccentricity, and to exhaust as they pass the line of greatest eccentricity of the drums. The steam is cut off at any desired point, and

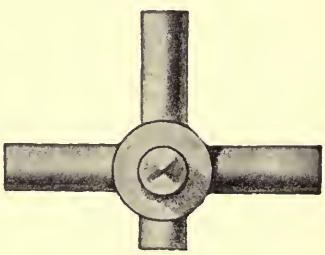


FIG. 240.

expands, while the piston in front of it is impelled forward at a greater velocity than that following it. The intervening space increases until the point of greatest eccentricity is reached, when the exhaust of steam between that pair of pistons takes place through the ports C′ in the drums. Each succeeding piston is acted upon in like manner, and a continuous rotative motion is thus imparted to the crank shaft, there being no dead point, and all parts are in perfect equilibrium. It must be clearly understood that the drums are fixed into the brackets, while the casing is in no way connected either to the crank shaft or to the pistons, but is free to rotate upon the drums and accommodate itself to the friction within the engine. Each cover of the casing has cast upon it a sleeve, which is carried along around the drum into a stuffing box in the bracket, and those sleeves form the bearings upon which the casing revolves. The sole plate is so made that steam entering at branch pipe S (see Fig. 239), to which the governor is fixed, passes up a steam way formed in bracket X, also along one side of the sole plate to bracket X′, and so is admitted into the interior of the casing through both drums at the same time.

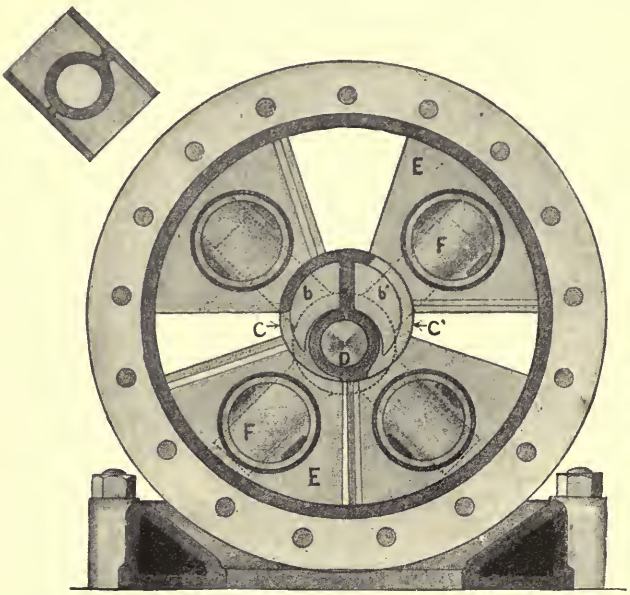


Fig. 241.—Pistons of Rota Engine.

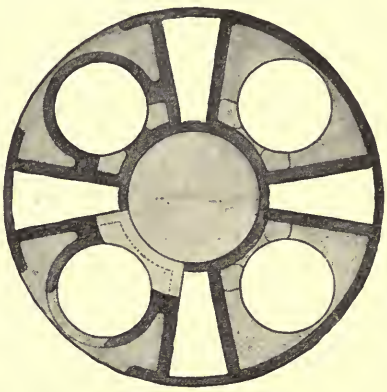


Fig. 242.—Casting of Pistons.

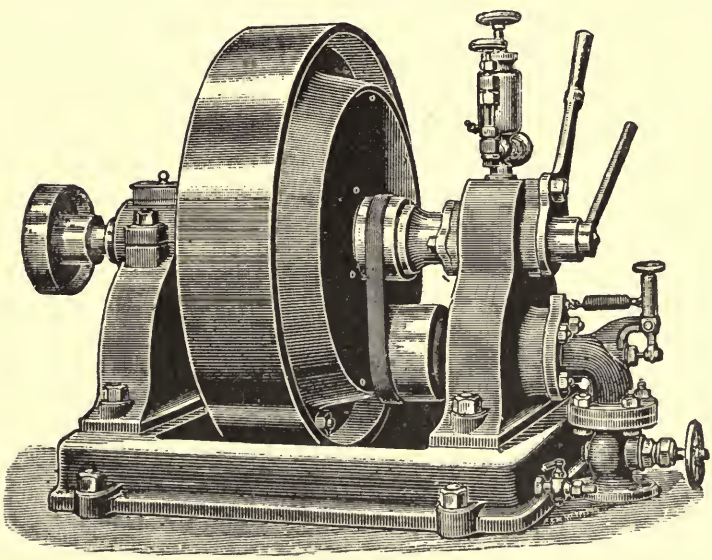


Fig. 243.—Rota Engine by Macewan Ross.

Similarly the exhaust steam passes down both brackets, and exhausts at branch T.

It will be seen from Fig. 241 that the steam chamber between the bottom pair of pistons is just opening to steam, and the chamber at the top is opening to exhaust.

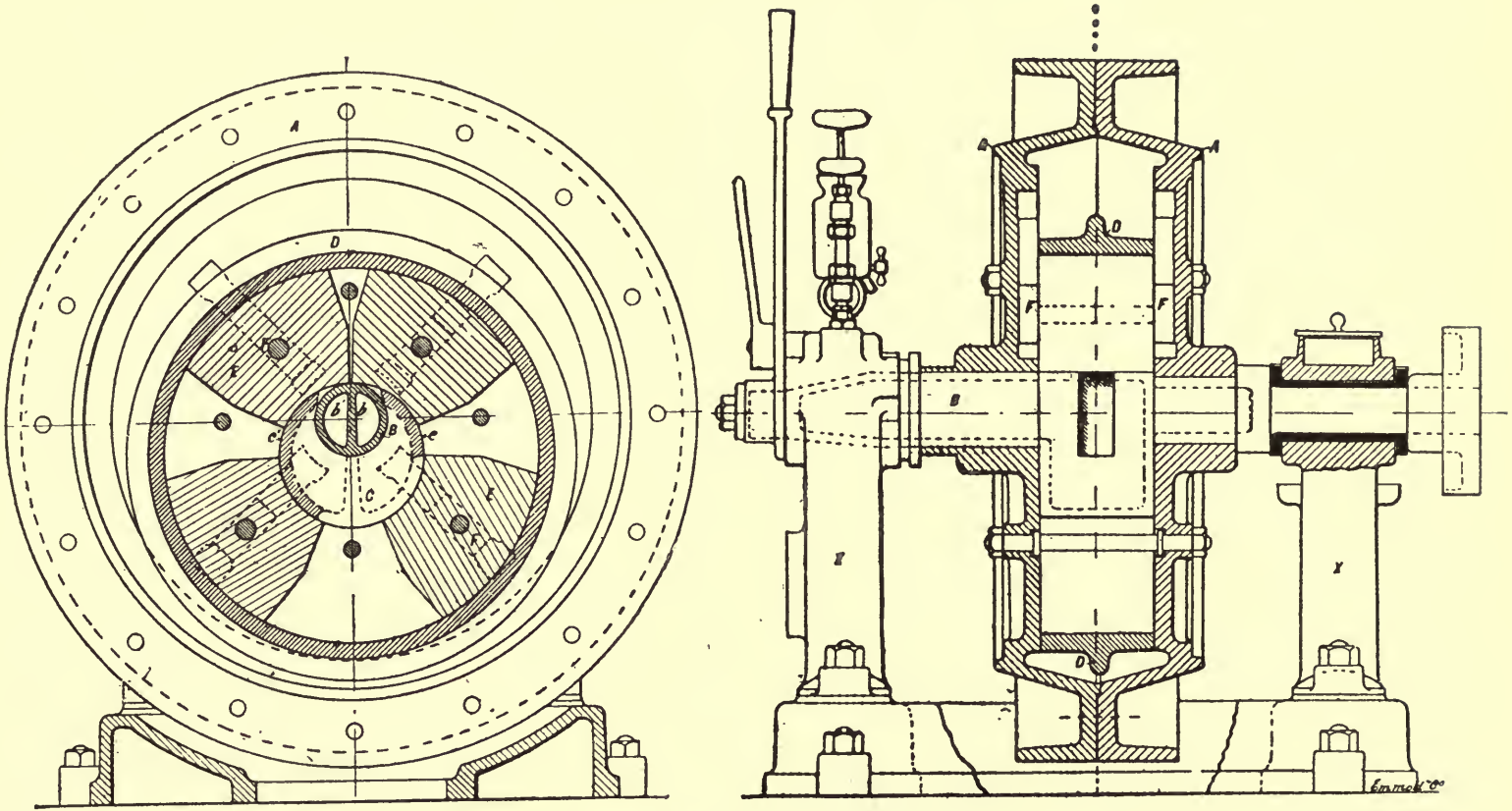


FIG. 244.—Macewan Ross Rota Engine.

In order to describe more clearly the cycle, suppose the middle of a side chamber to be full of steam, then the pressure would act against both pistons equally; but you will observe that the upper crank is much longer than the lower one, so that the upper crank will turn in an upward direction, taking the others round with it, until the chamber reaches the top centre, when exhaust will take place through port C′; and meanwhile the bottom chamber will be round to half-stroke and getting full steam, so that there are always two cranks doing work at the same time, and thus there is no dead point. The pistons therefore act as their own valves (the ports being in the fixed drums), so that the chambers only take what steam they require, and leave no passages full of steam after cut off takes place.

There is a steam space formed in the outside of each piston, so that the accumulated steam pressure acting between them and the casing is made almost to balance the centrifugal force of the pistons. It must also be noted that there is no *side* pressure on the pistons, and therefore if fitted accurately at first they should wear well.

This accuracy is easily attained, as all the work is done on the lathe, the pistons being cast all together, as shown on Fig. 242, with distance pieces between each, and are not separated until they have been turned and fitted to the drum, casing, and covers. Each piston and crank arm is fitted with suitable springs, those in the cranks being so arranged as to mutually accommodate themselves should there be any end movement of the shaft. Let us now consider some of the results which are produced when the engine is working. The pistons, which are close together when in the bottom position, are 3 inches apart when at the top, which is 3 inches travel in half a revolution, or a speed of 6 inches per revolution.

Now, suppose the engine to be running at its normal speed—800 revolutions per minute—the relative speed of piston is $800 \times .5$ feet $= 400$ feet per minute, and as there are four steam spaces we have a total "outward" stroke in each revolution of 12 inches, and yet the speed of piston is only 400 feet per minute.

The speeds of the various surfaces working in contact are also very low, as will be seen from the adjoining table:—

	Pressures on Surfaces working in contact, in Lbs. per Square Inch.	Speeds of Surfaces working in contact, in Feet per Minute.
Outside of pistons and casing . .	50	283
Side blocks and crank arms . .	435	266
Crank shaft journals . . .	78	366
Casing journals on drums . . .	40	900

Although the latter speed, namely, the bearings of the casing, seems high, still it is under constant pressure, and is well lubricated from holes being bored through into the exhaust passage *b′* in the drum.

The novel principle in this engine was the cylinder revolving with the pistons. Fig. 243 is a view of the complete engine with the shaft fitted with a coupling.

THE HULT ENGINE

Starting from the same point as Macewan Ross, but some years later, this inventor made some improvements on the "Rota" engine type, and by introducing roller and ball bearings reduced the friction.

Starting from this point, the inventors of the Hult engine have made the cylinder participate in the rotary motion of the piston, thus substituting rolling friction for the sliding friction, which had shown itself fatal to efficiency in other rotary engines, while

they are thereby enabled to readjust the contacts between the piston and the cylinder; these advantages belong to no other system.

PRINCIPLES OF THE HULT SYSTEM

The principles of the Hult engines are shown in the schematic Figs. 245 and 246. In order to simplify this description of the principles, only one shutter or door is shown, instead of two or three shutters placed at respectively 180° and 120°, with their steam passages for each shutter; for some purposes as many as six shutters have been placed in the same piston.

The piston F, placed eccentrically within the cylinder C, is shrunk on the main shaft A, and is provided with passages for the admission of steam H and for the exhaust I; the sliding shutter G is placed near the admission passage.

The steam from the boiler enters through the hollow shaft A and the passage H into the space enclosed between the piston, the cylinder, and the shutter, and pushes

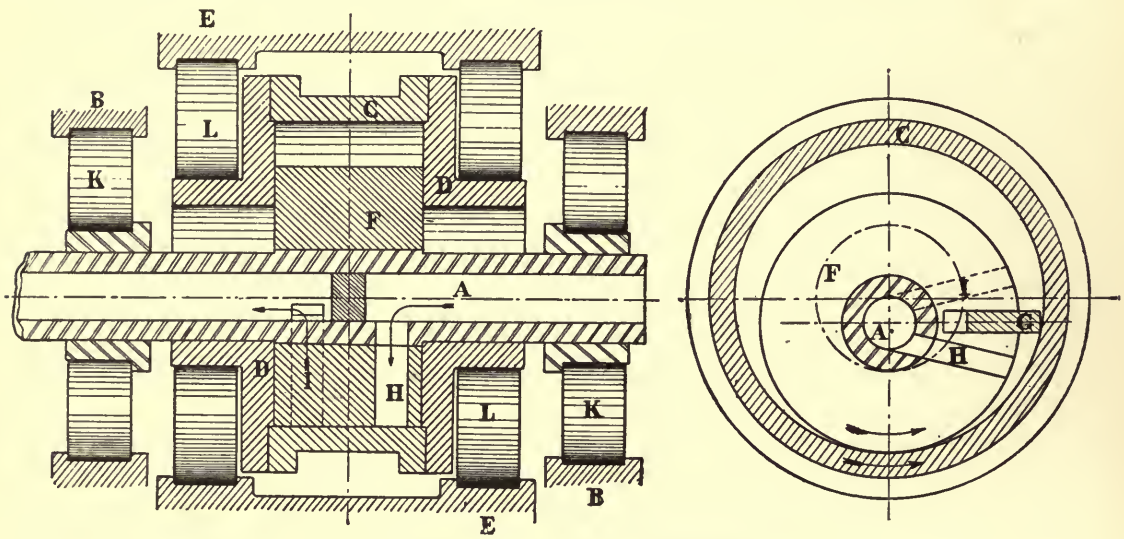


FIG. 245.—Single-Door Hult Engine.

round the shutter for turning the piston, the shutter having the same length as the piston.

When the piston reaches a certain angle the admission of steam is cut off automatically by a valve placed inside or at one end of the hollow shaft. The volume of steam introduced then works by expansion upon the shutter G, until the pressure becomes low enough, when the rotary motion of the piston, eccentric to the cylinder, uncovers the openings for the exhaust steam. There is no organ for admission or exhaust of steam in the cylinder itself.

Centrifugal force acting upon the shutter ensures a perfectly steam-tight fit between the shutter in the piston and the cylinder, both revolving in the same direction.

The shaft with the piston runs on the roller bearings B, K. The cylinder runs quite free on the roller bearings E, L; it is drawn round by the piston in consequence of the pressure between them at the line of contact. The piston and the cylinder have thus the same circumferential speed, but in consequence of their eccentricity the angular speed of the piston is slightly higher.

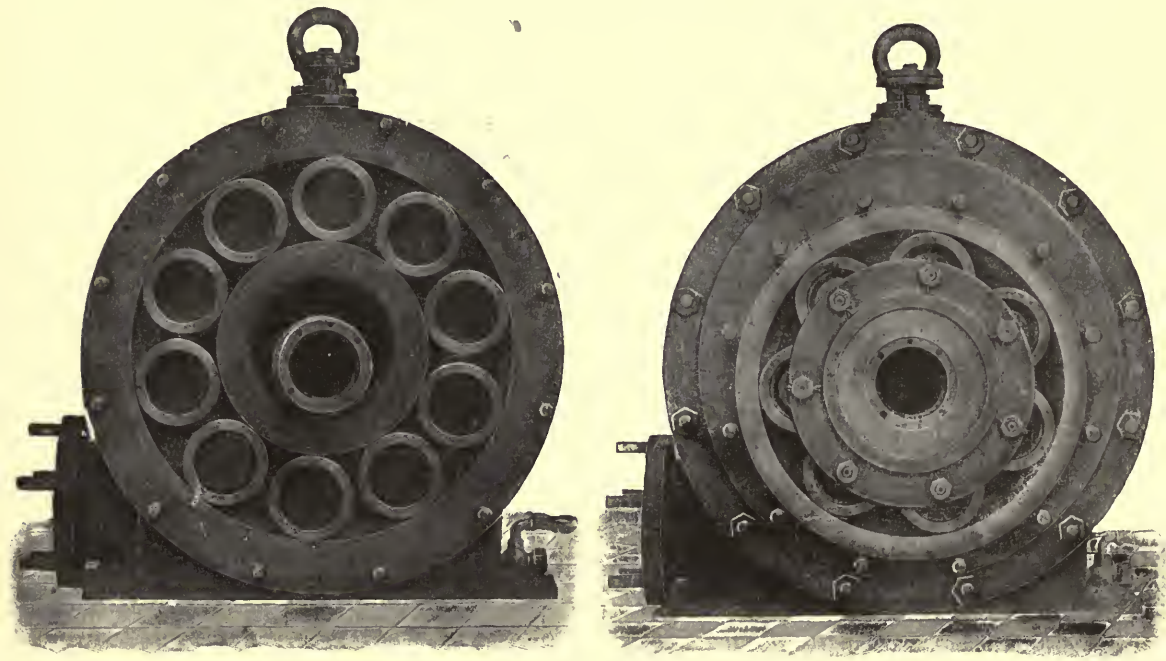
In the Hult engines with two shutters, placed at 180° to one another, as shown in Figs. 247 and 248, there are two periods of steam admission during each revolution,

and in the engines with three shutters, shown in Fig. 250, there are three such periods. The greater number of shutters increases the power of the engine, while reducing the condensation within the cylinder.

As the piston and the cylinder both revolve uninterruptedly in the same direction, the piston speed can be conveniently kept up to 50 or 60 feet per second, or about twice the speed of reciprocating engines.

The cylinder bearings and those supporting the shaft and piston are all roller bearings with tempered steel rollers, kept at their relative distances by guide rollers, as shown in Fig. 246; the rollers run on racers fixed in the frame and frame covers respectively.

When easing the nuts holding the frame covers the adjusting screws on the top



Bearings of Cylinder. Bearings of Shaft.

FIG. 246.—Roller Bearings of Hult Engine.

of the frame can press the piston and the cylinder together at the line of contact, so as to ensure that the cylinder follows the rotation of the piston, while keeping the line of contact steam-tight. The ends of the piston are also adjustable in relation to the cylinder, as described farther on.

Referring to Figs. 247 and 248, the following is an index to the parts :—

1. Cylinder.
2. Cylinder end on governor side.
3. Cylinder end on pulley side.
4. Adjustment disc for piston ends.
5. Adjustment wedges for piston ends.
6. Adjustment screws for piston ends.
7. Piston.
8. Sliding shutters.
9. } Steam admission passages.
10. }
11. Exhaust passages.
12. Exhaust orifices in piston.
13. Main shaft.
14. Frame.
15. Frame cover on governor side.
16. Frame cover on pulley side.
17. } Adjustment screws for line of contact between
18. } piston and cylinder.
19. Cover screw for reaching adjustment screws 6.
20. } Covers for shaft roller bearings.
21. }
22. Inner racer for cylinder roller bearing.
23. Outer racer for cylinder roller bearing.
24. Inner racer for shaft roller bearing.

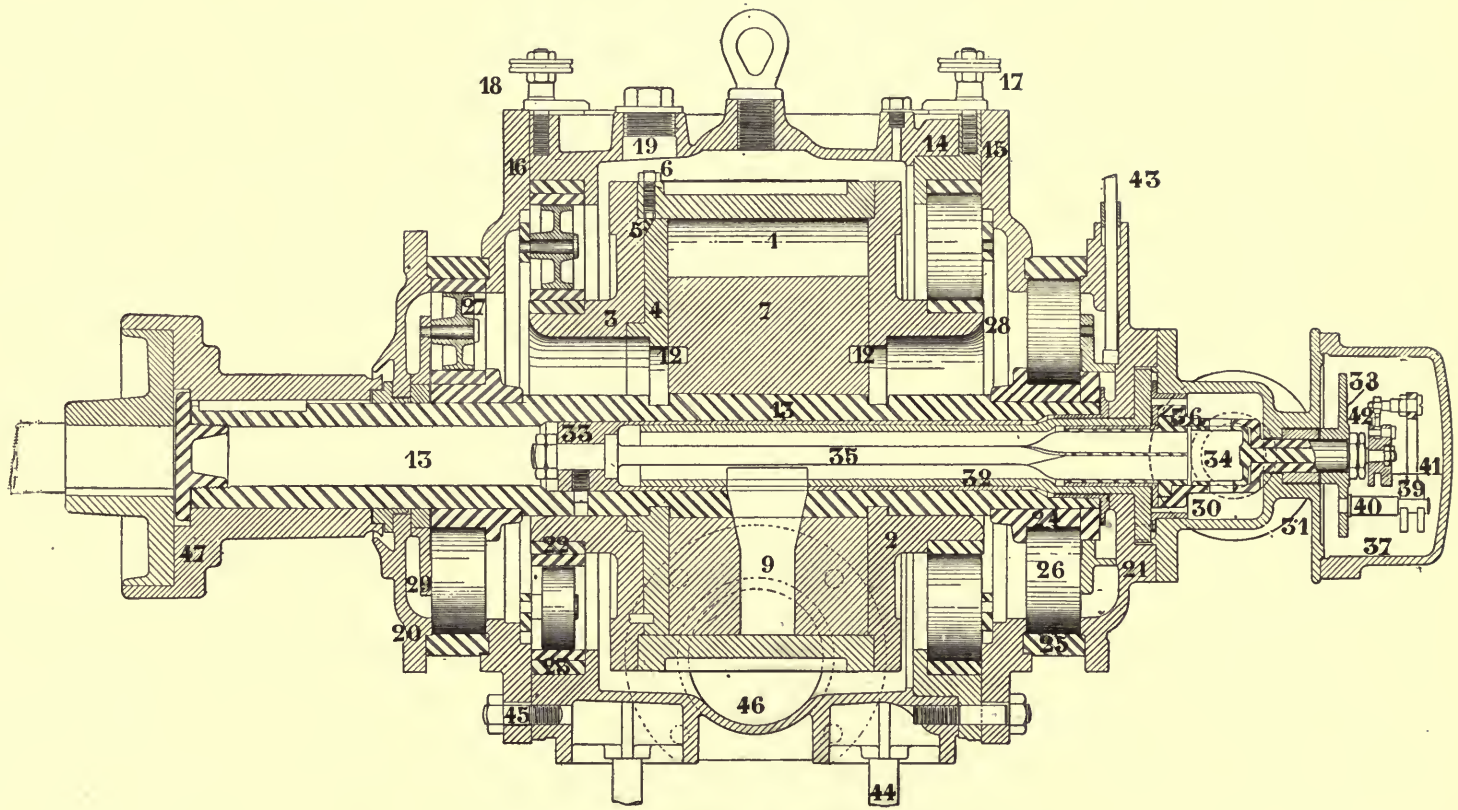


FIG. 247.—Section of Hult Engine.

25. Outer racer for shaft roller bearing.
26. Steel rollers in bearings.
27. Guide rollers.
28. Hoop for cylinder guide rollers.
29. Hoop for shaft guide rollers.
30. Steam admission box.
31. Flange for admission box.
32. Fixed steam distributing tube.
33. Revolving steam distributing tube.
34. Governor valve.
35. Connecting-rod between revolving distributing tube and admission valve.
36. Revolving steam admission valve.
37. Governor box.
38. Governor disc.
39. Governor weights.
40. Governor pivot.
41. Governor lever plate.
42. Governor lever.
43. Rod for turning automatic lubricator.
44. Holding-down bolts.
45. Cover bolts.
46. Exhaust pipe.
47. Coupling.

THE HULT TWO-SHUTTER ENGINE (FIGS. 247, 248, AND 249)

The steam from the boiler enters into the box 30, passes through tube 32, and reaches the cylinder through passages 9 and 10, every time one of them has passed the line of contact between the cylinder 1 and the piston 7. The steam acts on the sliding shutter 8, at first with full pressure, until the moment of the cut-off, and then by expansion.

The distributing arrangement, placed in the hollow main shaft 13, consists of an inner fixed tube 32, an outer revolving tube 33, and a rod 35, connecting the tube 33 with the admission valve 36, which is perforated by a number of slots, through which the steam can enter into the fixed tube 32. The tube 33, the rod 35, and the valve 36 thus participate in the rotary motion of the piston 7. The revolving tube 33 is pierced with slots opposite to the passages 9 and 10, so that steam at full pressure is admitted into the cylinder through these passages every time any of the slots passes the corresponding orifice in the fixed tube 32.

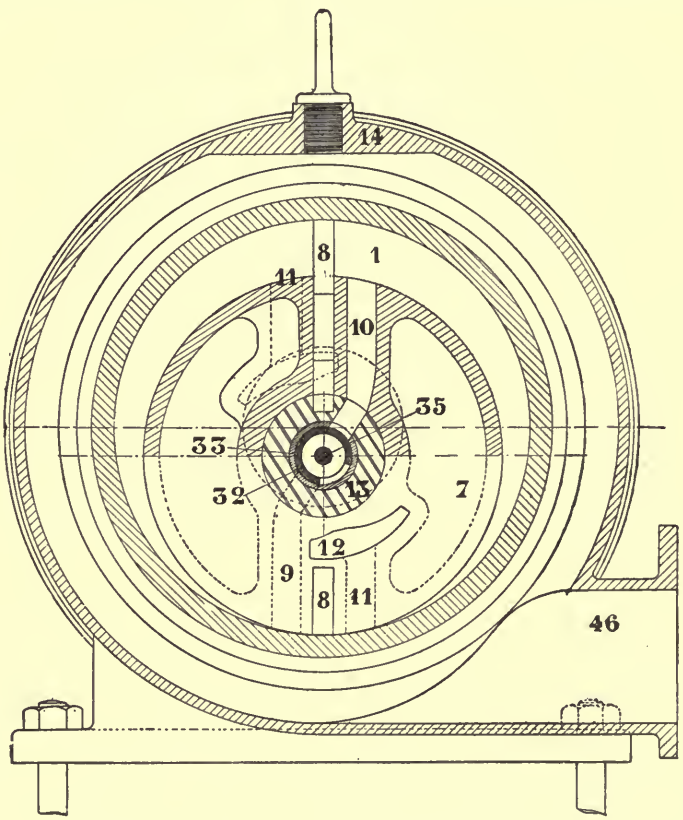


FIG. 248.—Steam Ports. Double-Door Section.

The governor, placed in the box 37, is of a centrifugal type; its disc 38, fixed to the steam admission valve 36, at the end of the rod 35, which is bevelled so as to allow steam to pass, turns at the same speed as the piston 7. The weights 39, equilibrated by a spring and mounted on the pivot 40 on the disc 38, are connected by levers 42 with the plate 41 on the rod of the governor valve 34, which has openings

corresponding to those on the admission valve 36. The weights 39, when separated by centrifugal force, pull at the levers 42, and give to the governor valve 34 an additional rotary motion, independent of that of the admission valve 36.

The expanded steam escapes from the cylinder through the passages 11 and the orifices 12 on the end faces of the piston.

In consequence of the eccentricity of the piston in relation to the cylinder there are open spaces inside the end pieces of the cylinder to allow for the movement of the shaft. As the position of the transversal surfaces of these spaces does not change during the rotation of the engine, the escapement periods are determined by the shape and section of the orifices 12.

The exhaust steam then spreads throughout the interior of the frame 14, and issues finally through the exhaust pipe 46 to the condenser, or into the outer air.

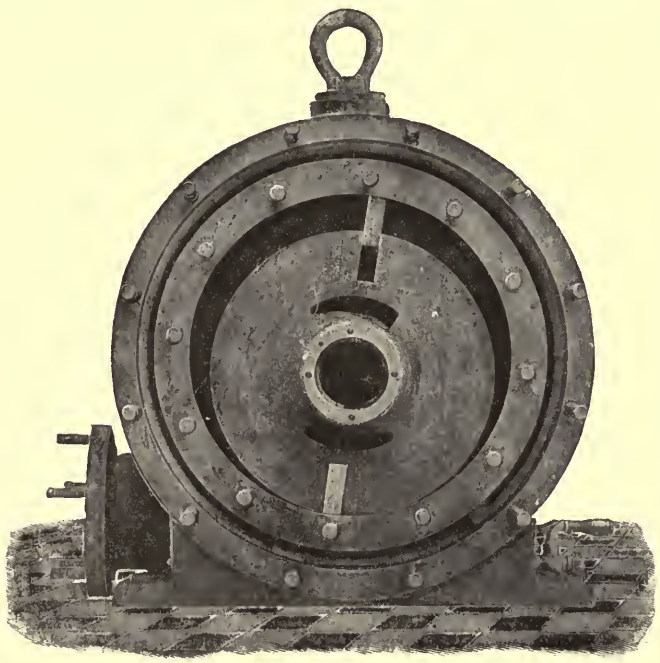


FIG. 249.—Steam Ports.

The roller bearings for the cylinder consist of tempered steel rollers 26, running on steel racers 22, fitted on the annular parts 2 and 3 of the two cylinder ends. The distance between the rollers is kept constant by guide rollers 27, mounted on pins fixed on hoops 28. Two other steel racers 23 are fitted in the frame 14, outside the rollers concentrically to the inner racers 22.

The driving shaft 13, carrying the piston 7, is supported by other similar bearings, consisting of the rollers 26, the inner racers 24, guide rollers 27, hoops 29, and outer racers 25.

These shaft bearings, independent of the cylinder bearings, can be adjusted vertically by turning the adjustment screws 17 and 18, which give to the piston 7 the exact pressure against the cylinder necessary for rotating the cylinder, and for ensuring a perfectly steam-tight fit at the line of contact.

A disc 4 is placed at the left end of the piston for adjusting a suitable steam-tight fit between the ends of the piston and the cylinder; the position of this disc is modified by the adjustment screws 6 pressing upon the wedges 5, which screws are reached by a screwdriver on removing the cover screw 19.

Whatever slight friction there may be at the ends of the piston is almost inappreciable, as the cylinder rolls freely in the same direction as the piston, and with but slight difference in the angular speed, while their circumferential speed is the same.

The question of lubrication is important for all high speed engines. The Hult engine is easily lubricated, because rolling friction is almost entirely substituted for the sliding friction existing in all reciprocating engines; the steam on entering the engine takes up the exact quantity of lubricating oil required from one single automatic lubricator driven by the rod 43, set in motion by an eccentric cam on one of the hoops 29.

THE HULT THREE-SHUTTER ENGINE (Fig. 250)

This is in most respects similar to the two-shutter engine, except that there are three shutters, with three admissions of steam, and the cycle of admission, expansion, and expulsion of steam for each shutter is completed during one revolution of the shaft.

When a three-shutter piston with its arrangements for admission of steam is substituted for a two-shutter piston in the same engine the power is increased, and it becomes perhaps even better equilibrated than before; while the condensation of steam in the cylinder becomes, if possible, even less detrimental.

When the exhaust commences of the steam, which has acted upon one of the shutters, the next shutter is driven by expanding steam, and the third shutter by steam at full pressure; while immediately afterwards, and before the said exhaust is completed, fresh steam is again let on to the first shutter.

Fig. 250 shows in transverse section the steam admission arrangement of a three-shutter engine, of the same size as that of which a longitudinal section is given in Fig. 247.

The three shutters are marked 8, the three admission passages 10, and the three exhaust passages 11; the steam admitted for one shutter through the passage 10A acts upon the shutter 8A, and is exhausted through the passage 11A; and in the same manner 10B, 8B, and 11B, as well as 10C, 8C, and 11C, co-operate together.

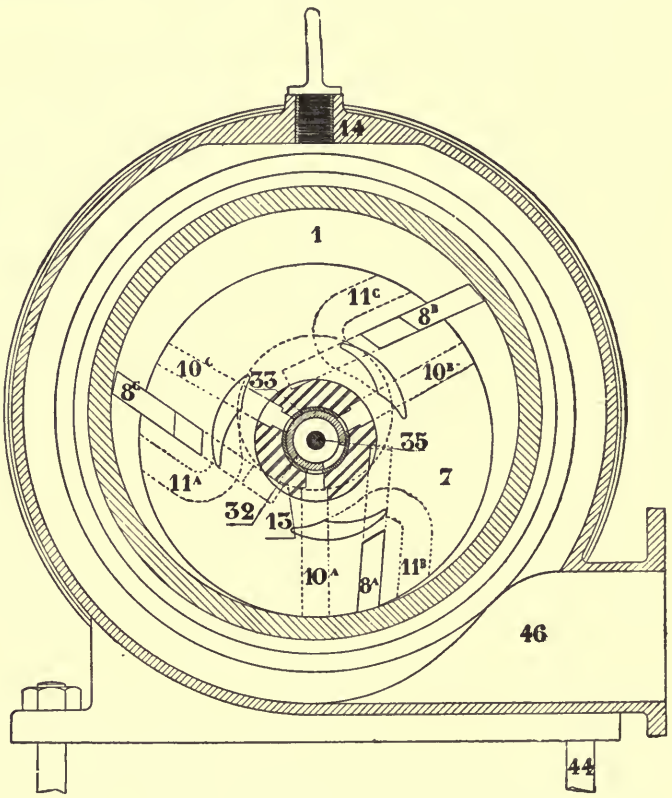


Fig. 250.—Three-Door Hult Engine.

THE HULT REVERSIBLE ENGINE FOR MARINE PURPOSES, AUTOMOBILES, ETC.

Fig. 251, representing on one-tenth scale a Hult marine engine giving 45 brake horse-power at 10 atmospheres pressure, shows its convenient form; the 200 brake horse-power occupies over all a space 6 feet long, 4 feet high, and 3 feet wide. The weight of the marine engines generally vary from 30 lbs. to 45 lbs. per brake horse-power, according to power and steam pressure.

Engines for these purposes must be light and occupy small space; they must be simple in construction, easy to manage, and handy, so that all changes of speed and direction can be executed easily and quickly.

In spite of all recent improvements, reciprocating engines do not fulfil these

conditions as perfectly as Hult engines, which work directly on the screw shaft with higher speed than can be obtained with reciprocating engines, while their small dimensions are in every way convenient.

Being free from shocks and vibration, they can be mounted in ships and boats of light construction; as the engine shaft is coupled in direct prolongation of the screw shaft, without cranks or fly-wheel, their centre of gravity can be placed much lower than with reciprocating engines.

The description given above of stationary Hult engines also applies to the marine engines, except in the arrangements for the distribution of steam, which in the latter are modified to suit the requirements for reversion of direction and variation of speed.

The reversion is obtained by moving lengthwise the distributing tube for admitting the steam on one side or the other of the sliding shutters.

Fig. 251 is a section of the Hult marine engine in the position of driving "forward" at full speed. The distributing tube C, which does not participate in the rotary motion of the main shaft A, is moved lengthwise by a lever D, connected by a rod E with a fixed point P on the frame.

The steam introduced from the boiler into the box F passes through the orifices G into the distributing tube C, pierced by the slots H, which by the longitudinal movement of the tube can be brought in line with one or the other of the steam passages M and N, as required for the desired direction of rotation.

As the tube C is closed at one end by the stop I, the steam can only reach the cylinder through the slots H; the exhaust steam escapes through the orifices K or L.

In the position shown in Fig. 251 the slot H opens fully into the steam passage M, admitting steam for driving "forward"; the exhaust steam passes through the steam passage N and (as soon as this passage reaches the orifice K) into the space outside stop I, in the distributing tube; it then passes through O into the space outside the cylinder, and finally to the exhaust pipe R.

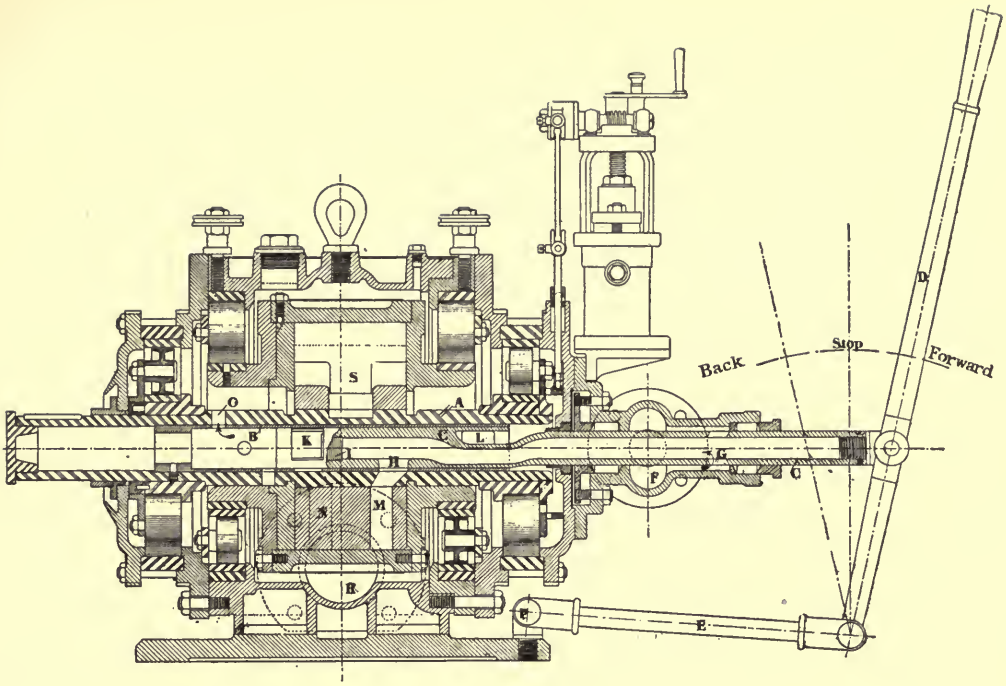
When the lever D is set for "back," the slots H come opposite the steam passage N for admission of steam; the other passage M serving in this case for the exhaust each time it opens into the orifice L, cut in the distributing tube, which is there flattened out for the passage of the exhaust steam.

This single lever thus ensures the perfect reversibility of the engine at all positions of the piston, and the variation of speed depends upon how far the lever D is moved from its central "stop" position in the direction of the "forward" or "back" positions.

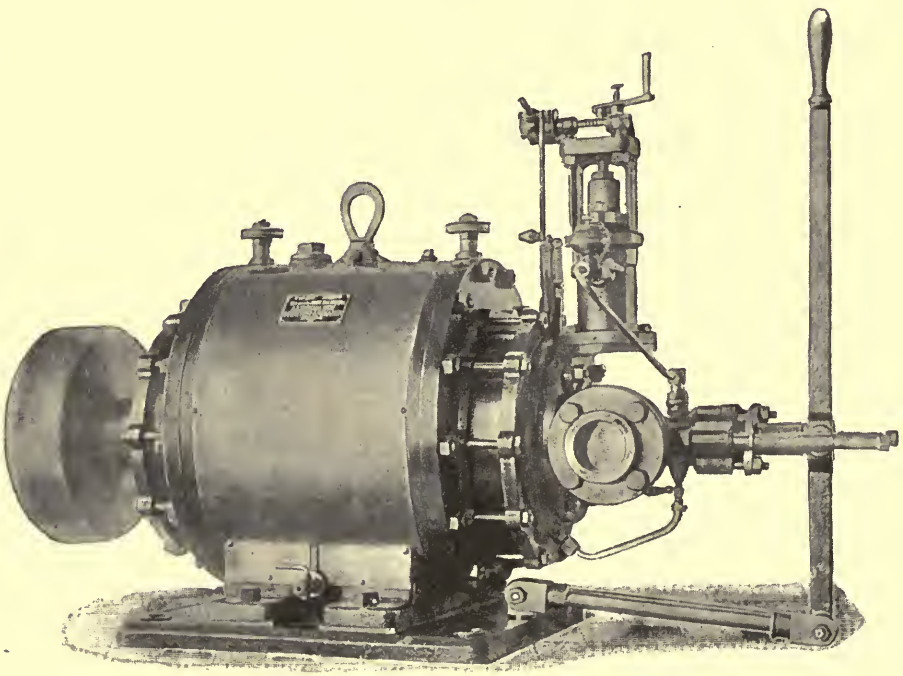
The following table gives the approximate power, speed, and weight of the three-shutter engines, as well as their consumption of dry steam *per brake horse-power* with *exhaust into open air.*

TABLE XXVII.

142 Lbs. Pressure.		Number of Revolutions per Minute.	Weight of Complete Engine.
Brake Horse-Power.	Consumption of Steam per Brake Horse-power and Hour, with Exhaust into Open Air.		
	Lbs.		Cwt.
8	...	1500	1¼
12	36	1300	4½
20	33	1250	6
40	31	1000	12
60	29½	850	17½
80	28	700	23
120	27½	550	38
200	27	500	52



Sectional View.



Complete View.

FIG. 251.—Hult Rotary Marine Engine.

The above figures of consumption of steam without vacuum are, of course, reduced when working with condensation, with the usual ratio of gain depending upon the vacuum given by different types of condensers.

In comparing these figures with those of other engines it must not be overlooked that the figures here given are for brake horse-power; if stated for indicated horse-power, as is sometimes done, the figures would be proportionately lower.

This chapter shows the best that has hitherto been done in the direction of the Rotary Piston Engine. It may be observed that one objection still remains, and that is the heavy strains in other directions than that upon the shutter, and not in the direction of motion. The push is not direct in the line of motion, but constrained into that line by the cylinder.

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